The
ENGINEERS' DIGEST

REVIEW OF ENGINEERING PROGRESS ABROAD

OL. IX. No. 2 FEBRUARY, 1948

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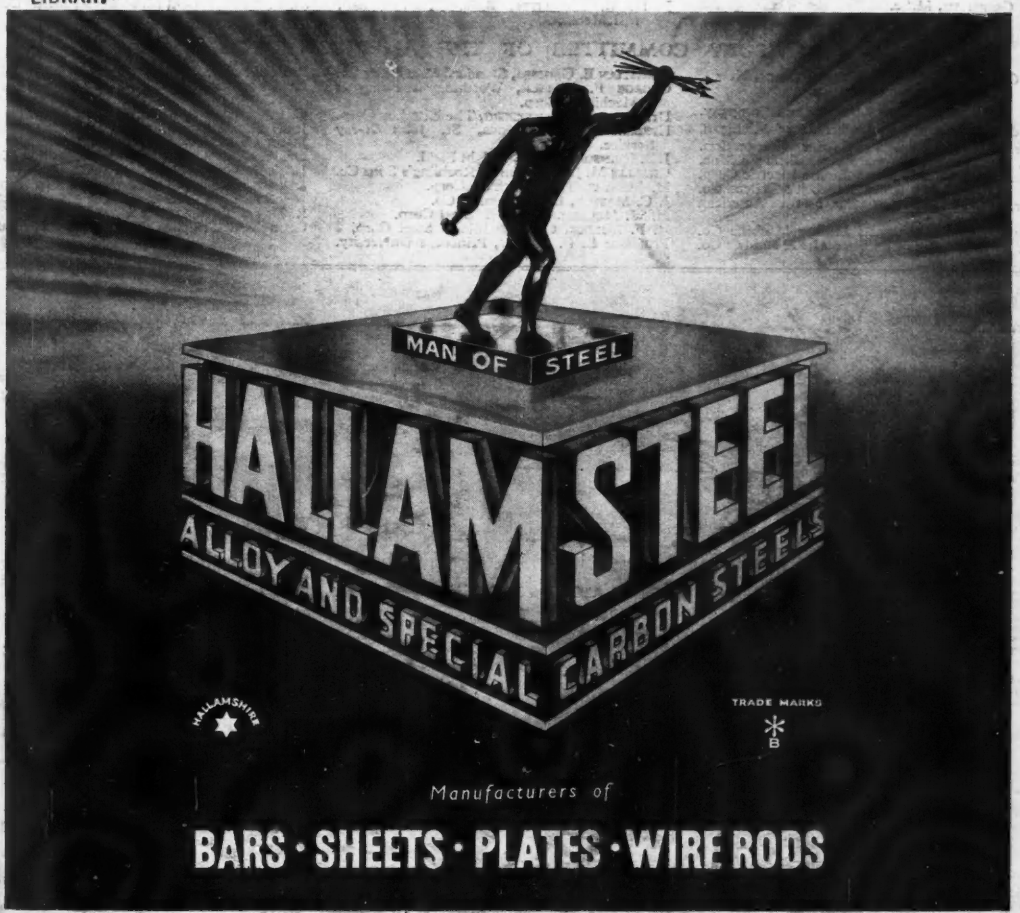

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THE ENGINEERS' DIGEST

Published Monthly by Engineers' Digest Ltd., 120 Wigmore Street, London, W.1. Tel.: WELBECK 9357

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Contributions to the Theory of Cyclone Dust Collectors

By F. SCHULZ. (From *Maschinenbau und Wärmewirtschaft*, Vol. 1, No. 5/6, November/December, 1946, pp. 89-100, 17 illustrations.)

CONSIDERABLE attention is being given to the development of cyclones both in the U.S.A. and in Great Britain, and some interesting articles on this subject have recently been published in the technical press of these countries. Articles such as *How the Cyclone Works on Mineral Separations* giving a description of the Driessen cyclone under test in the U.S.A. (*Engineering and Mining Journal*, McGraw-Hill, Vol. 148, No. 12, Dec., 1947) and *Centrifugal Dust Collectors* (H. L. Muntgomery Larcombe, *The Mining Magazine*, Vol. 77, Nos. 3, 4, 5 and 6, 1947) deal elaborately with the performance and design of cyclones; the results obtained, however, are based on empirical deduction. E. Feifel's new conception, being rather unconventional in its approach to the theoretical problems involved, is the subject matter of the present article which shows how flow patterns in cyclones can be analysed mathematically and clearly demonstrates the influence of relative dimensions upon efficiency. Attention has been drawn to this remarkable development in a previous issue of *The Engineers' Digest* (see E. Feifel, *The Cyclone Dust Separator*, November, 1947, p. 368).

AMONG the dust collectors used in recent years, the cyclone with cylindrical separating chamber has become of increasing importance. Essentially it consists of a small duct with tangential entry, a concentric outlet duct, a peripheral dust outlet slot, and a central hole in the plate bottom (Fig. 1).

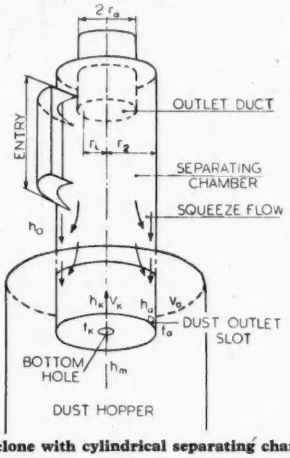

Fig. 1. Cyclone with cylindrical separating chamber.

The theory of cyclones has repeatedly been dealt with in the relevant literature[1, 2, 3]. Yet there is a large number of problems still to be solved, these problems being due mainly to the intricacy of the flow pattern.

I. THE VACUUM IN THE SEPARATING CHAMBER AND ITS INFLUENCE ON PERFORMANCE.

One of the basic conditions for efficient separation is a well developed squeeze-flow which should rapidly take the dust from the entry zone of the separator and deposit it in the dust hopper. On leaving the separating chamber, the dust path is dependent on the constructional features of the collector, which can be of one of the following types :—

1. Separating chamber entering dust hopper without making use of a plate bottom.
2. Separating chamber and dust hopper divided by a plate bottom leaving an annular slot between both spaces.
3. Dust outlet reduced to a relatively narrow slot.

The dimensions and proportions of the slots in case 2 and case 3 depend on the pressure drop desired. In case 1, no pressure drop obtains as there is an unrestricted cross-section between separating chamber and dust hopper. Obviously, cases 1 and 2 are limiting cases to case 3.

The pressure drop Δh_a is dependent on the mean pressure head h_m in the dust hopper which is a function of the pressure head h_a in front of the dust outlet slot and that behind the bottom hole h_k, as well as a function of the slot and bottom hole areas :—

$$\Delta h_a = h_a - h_m \quad .. \quad .. \quad .. \quad (1)$$

The pressure heads are measured in metres gas column (for a gas weight per unit volume γ_1 we have $h = p/\gamma_1$ meters).

Assuming frictionless fluid, the maximum values

$$h_a = h_0 - \frac{c_e^2}{2g} \text{ and } h_k = h_0 - 2h_e \quad .. \quad (2)$$

are determined by the inlet velocity c_e of the cyclone, and its velocity head $h_e = \frac{c_i^2}{2g}$, where c_i is the velocity at the radius r_i.

Thus, h_m remains to be calculated. Because of the small pressure deviation of h_a and h_k from atmospheric pressure (about 0·01 to 0·04 atmospheres), isothermal treatment of the problem will suffice.

The condition of continuity requires the dust outlet slot and the bottom hole to pass equal quantities of air.

$$v_a f_a = v_k f_k \quad .. \quad .. \quad .. \quad (3)$$

The velocities v_a and v_k are dependent on the pressure difference between the separating chamber and the dust hopper, and separating chamber and bottom hole respectively.

$$v_a = \zeta_a \sqrt{2g(h_a - h_m)}$$

and

$$v_k = \zeta_k \sqrt{2g(h_m - h_k)}$$

The coefficients ξ_a and ξ_k take account of shape and dimensions of the dust outlet slot and bottom hole. From the expressions for v_a and v_k, and using eq. (3) the mean pressure head in the dust collecting container can be found :—

$$h_m = \frac{h_a + h_k \left(\frac{\zeta_k f_k}{\zeta_a f_a}\right)^2}{1 + \left(\frac{\zeta_k f_k}{\zeta_a f_a}\right)^2} \quad .. \quad .. \quad (4)$$

The coefficients ζ_a and ζ_k are difficult to assess numerically, but the approximation $\zeta_a = \zeta_k$ appears to be justified.

By denoting the ratio $f_k/f_a = n$, we have

$$h_m = \frac{h_a + h_k n^2}{1 + n^2} \quad .. \quad .. \quad (5)$$

Therefore, the pressure drop according to eq. (1) can be written as

$$\Delta h_a = (h_a - h_k) \frac{n^2}{1 + n^2} \quad .. \quad (6)$$

and, by using eq. (2), its maximum value is found as

$$\Delta h_{a\,max} = \left(2\,h_e - \frac{c_e^2}{2g}\right)\frac{n^2}{1+n^2} \quad .. \quad (6a)$$

For the calculation of the air speed v_k the pressure drop through the bottom hole will be of importance

$$\Delta h_m = h_m - h_k$$

By substituting for h_m from eq. (5) we obtain

$$\Delta h_m = (h_a - h_k)\frac{1}{1+n^2} \quad .. \quad .. \quad (7)$$

or $$\Delta h_{m\,max} = \left(2\,h_e - \frac{c_e^2}{2g}\right)\frac{1}{1+n^2} \quad .. \quad (7a)$$

Although eq. (6) expresses the pressure drop as a function of the ratio of the bottom hole and slot areas, it gives no indication of the influence of the actual dimensions of these areas. Also, eq. (7a) does not establish a relationship between pressure drop and distance of plate bottom from the inlet so that $\Delta h_{m\,max}$ as obtained by this formula is of theoretical interest only. To assess these influences, the pressure distribution at arbitary cross-sections of the separating chamber must either be known or be determined experimentally. Fig. 2 shows such pressure distributions at the plate bottom of the cyclone for three arbitary positions (I to III) of the bottom, $z = B$, $z = 6{\cdot}25B$ and $z = 11{\cdot}5B$ (B = = height of tangential entry, z = distance of plate bottom from outlet duct which, in this case, coincides with the upper edge of the inlet). The vacuum is given in millimetres water column (gauge pressure).

The displacement of the vacuum peak ϵ from the geometrical cyclone axis is due to the asymmetry of the inlet. For a uniformly distributed entry—e.g. guide vanes in a turbine—the pressure distribution would be symmetrical with respect to the cyclone axis (curve a in Fig. 4). From the tests represented in Fig. 2, it has been found that $\epsilon = \frac{r_a}{2}$. The displacement of the pressure distribution curve a in Fig. 4 by ϵ from the centre (curve b) implies that h_a does not remain constant around the circumference but varies between $h_{a\,max}$ and $h_{a\,min}$. Similarly, the absolute pressure along any parallel line to the cyclone axis does not change continuously but fluctuates about a mean curve. With increasing distance z from the bottom, the vacuum peak decreases (see Fig. 2) while the vacuum near the wall of the separating chamber increases, until, at a certain value of z, the vacuum is constant over the cross-section. It is of interest to observe the decrease of the velocity head h_e with increasing bottom distance z. Therefore, from the point of view of energy expenditure, the separating chamber should be made as long as possible.

Measurements and observations on a glass model are summarized in Fig. 3, where it can be seen that the vacuum peaks lie on a cylindrical helix with a diameter $2\,\epsilon$ (in Fig. 2, $2\,\epsilon = r_a$) and a constant pitch σ ($\sigma = 10{\cdot}5\,r_a$). Also the dust path is a cylindrical helix of constant pitch. Measurements can be made up to a plate distance of $4\,\sigma$ to $5\,\sigma$ because, up to this distance, the rotational flow along the chamber walls is stable. However, once this limit is exceeded, a breakdown of the flow with distinct radial components becomes apparent (Fig. 3). The helix pitch which, so far, was constant, suddenly decreases, and the dust which has not been deposited on the chamber walls, moves towards the bottom hole to be taken up by the upward core flow through the bottom hole (or core hole, as it is sometimes called). The inversion point is clearly

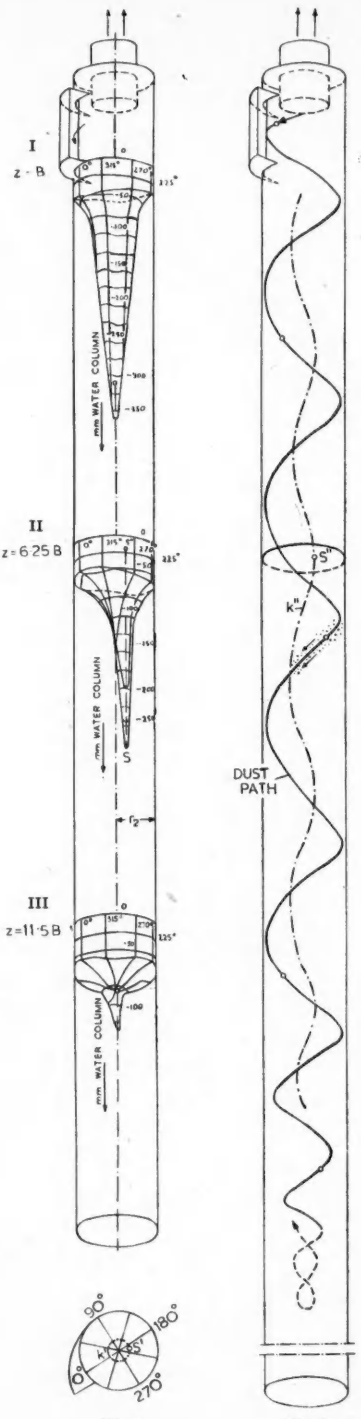

Fig. 2

Fig. 3

Fig. 2. Vacuum at the plate bottom.
k' is the plan view of the geometric locus of the vacuum peaks *S'*.

Fig. 3. Dust path and geometric locus of vacuum peaks.

discernible by a cork-screw-like air stream permeating the dust cloud (see dotted curve in Fig. 3).

According to these observations, the requirement for optimum separating efficiency would be a separating chamber without plate bottom, long enough to allow deposition of dust beyond the inversion point. This means that by shortening the chamber and by providing a plate bottom with concentric hole, the core flow would carry the undeposited part of the dust upward into the clean gas. In other words, a certain proportion of the dust removed through the peripheral dust outlet slot, is re-entrained in the carrier gas which is returned from the hopper into the separating chamber via the central bottom hole.

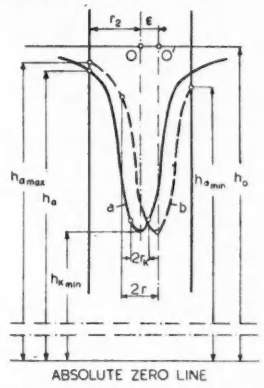

Fig. 4. **Pressure distribution at plate bottom.** Cyclone with entry evenly distributed (curve *a*), and with asymmetric (single) entry (curve *b*).

There is a definite answer to the question of optimum bottom distance, as regards keeping the pressure drop Δh_a as high as possible ; the smaller the bottom distance z the greater is the pressure drop Δh_a. How far this requirement can be reconciled with the aforementioned condition for optimum separating efficiency will be shown in Chapter II.

As mentioned before, the pressure drop Δh_a is a function of the radius r_k of the bottom hole. From Fig. 4 it can be seen that an increase of r_k reduces the vacuum at the hole. However, too small a hole, owing to its throttle action, has the same effect, i.e. a reduction of Δh_a. From this reasoning it appears that there must be an optimum hole size.

For direct sucking-off, the problem is simplified inasmuch as, by having no bottom hole, any desired

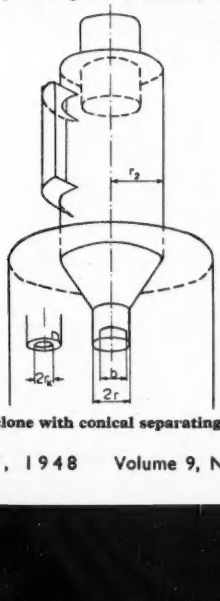

Fig. 5. **Cyclone with conical separating chamber.**

Δh_a can be obtained with corresponding energy expenditure on suction.

Although the tests described have been made with a view to obtaining the optimum pressure drop, they can be used for proving the inadequacy of many a conical cyclone such as shown in Fig. 5. A pressure plot around the circumference of the separating chamber bottom reveals a much greater variation in pressure than is the case with the cylindrical chamber.

Consequently, for maximum pressure drop, the position of the dust outlet slot with respect to clean gas entry is of great importance. On the other hand, the small difference between the pressures at outlet slot and bottom hole leads to the employment of a single peripheral slot.

The amalgamation of two functions corresponding to two separate openings endangers the stability of separation. Indeed, a small external impetus may transform the stable state into an unstable one. Such an impetus, for instance, may come from a small leakage in the dust hopper, and would eliminate the pressure difference which effects dust withdrawal. It is not only that separation may cease, but also that dust already deposited may be picked up again by the interference flow and returned into the clean gas outlet duct, a process eventually leading to the emptying of the dust hopper. Experiments have proved that this may happen with leakages too small to have the slightest effect on the collection efficiency of cylindrical chambers. With cylindrical separating chambers the leakage area may be equal to the dust outlet slot area without appreciably affecting the collection efficiency. From the foregoing it will be obvious why conical cyclones cannot be switched in parallel to a common dust hopper. Already differences in the manufacture of the inlet opening (admission) may suffice to make the separator ineffective.

In this connexion it may be of interest to investigate the maximum permissible leakage in the dust hopper, at which the pressure drop would still be positive thus depositing dust in the hopper, and at which the bottom hole would still be capable of consuming the leakage air. If f_u is the leakage area and v_u the velocity of the leakage air, we have

$$v_u = \zeta_u \sqrt{2g(h_o - h_m)}$$

and instead of eq. (3) we can write

$$v_k f_k = v_a f_a + v_u f_u \quad .. \quad .. \quad (8)$$

Assuming $\zeta_k = \zeta_u$ and substituting for the velocities v_k, v_u, and v_a we have

$$\frac{f_k}{f_u} = m = \frac{\sqrt{h_o - h_m}}{\sqrt{h_m - h_k} - \frac{1}{n}\sqrt{h_a - h_m}} \quad .. \quad (9)$$

After determining h_k and h_a by experiment, or calculation of extreme values, h_m can easily be obtained. Dust is being withdrawn as long as the pressure drop $\Delta h_a = h_a - h_m > 0$.

The theoretical border case is obtained when $h_m = h_a$ for $\Delta h_a = 0$. The maximum leakage area, which is independent of n becomes

$$f_{u\,max} = f_k \sqrt{\frac{h_a - h_k}{h_o - h_a}} \quad .. \quad .. \quad (10)$$

Tests on the cylindrical cyclone have shown that leakage areas as large as the dust outlet slot have no appreciable influence on the collection efficiency.

II. The Influence of the Vortex Root on the Bottom Distance.

In the foregoing it has been shown that the condition for optimum pressure drop Δh_a requires a shortest possible distance of bottom to inlet.

It is, possible, however, for every dust grain already in the hopper to re-enter the separating chamber with the rising air stream through the bottom hole. The effectiveness of the bottom hole would be utterly destroyed if the re-entering dust had no opportunity to reach the separator wall and again to be collected in the hopper. Every dust particle of the diameter a, once back in the separating chamber, can only be re-deposited, if it is given sufficient time to reach a circle of radius r_a before being raised to the outlet duct. As shown in Fig. 6, the particle will then be driven towards the wall by the rotational flow, and brought down by the squeeze-flow to be withdrawn through the dust outlet slot.

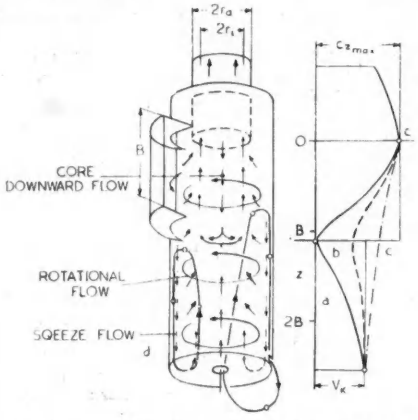

Fig. 6. Flow conditions in a cyclone with cylindrical separating chamber.

The minimum distance from the bottom to the outlet duct is governed by the minimum size of grain a_{min} [4] which can theoretically be removed.

$$a_{min} = \sqrt{18\,\nu\,\frac{\gamma_1\,c_r}{\gamma_s\,b_a}}$$

where $$c_r = \frac{1}{2\pi}\,\log_e\frac{r_2}{r_1}\,\sqrt{2g\,h_e\,\frac{r_i}{r_a}}$$

and $$b_a = 19\,h^{5/4}\,Q^{-1/2}$$

In solving the present problem, it is sufficient to define the position of the dust grain by two co-ordinates,

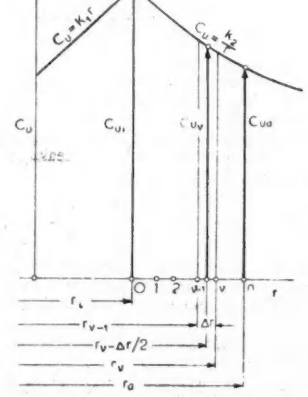

Fig. 7. Theoretical distribution of the peripheral velocity c_u in a cyclone.

i.e. the radial distance r and the distance z from the plate bottom.

To be able to calculate the rather complex process, certain simplifying assumptions must be made :—

1. The law of constant angular velocity should apply within the cylindrical space of radius r_i, whereas in the separating chamber (beyond r_i) the law of constant change of momentum is taken to hold good (Fig. 7).

Then

$$c_u = k_1 r \text{ for } r < r_i \quad .. \quad .. \quad (11)$$

and

$$r c_u = k_2 \text{ for } r > r_i \; .. \quad .. \quad .. \quad (12)$$

k_2 is obtained from the inlet conditions, and k_1 from equal peripheral velocity at r_i according to both laws.

$$k_1 = \frac{r_e\,c_e}{r_i^2} \quad .. \quad .. \quad .. \quad (11a)$$

$$k_2 = r_e\,c_e \quad .. \quad .. \quad .. \quad (12a)$$

2. The radial velocity of the air c_r within the radius r_a is negligible when compared with the relative velocity w_r of the dust grain, as, in the core zone, the radial flow is completely deflected into an axial one. Owing to its intertia, the dust particle does not follow this sudden change of direction.

Thus

$$c_r = 0 \text{ and } w_r = \frac{dr}{dt} \quad .. \quad .. \quad (13)$$

3. The smallness of a_{min} (abt. 10^{-4} cm.), permits Stokes's law of resistance to be applied :—

$$W = k_3\,w_r \quad .. \quad .. \quad .. \quad (14)$$

$$\text{with} \quad k_3 = 3\pi\nu\,\frac{\gamma_1}{g}\,a_{min}$$

With the above assumptions, the equation of motion in radial direction for a dust particle of the mass m the centrifugal force F, and the resistance W is given as

$$m\,\frac{d^2r}{dt^2} = F - W$$

and for r within r_i we have the differential equation

$$\frac{d^2r}{dt^2} + \frac{k_3}{m}\,\frac{dr}{dt} - k_1^2\,r = 0 \quad (0 < r < r_i) \; .. \quad (15)$$

the solution of which for the boundary conditions $t = 0$, $r = r_k$ (bottom hole radius) and $dr/dt = 0$ is

$$r = r_k\,\frac{-\lambda_2}{\lambda_1 - \lambda_2}\,e^{\lambda_1 t} + r_k\,\frac{\lambda_1}{\lambda_1 - \lambda_2}\,e^{\lambda_2 t} \quad (16)$$

Between r_i and r_a we have the equation of motion

$$\frac{d^2r}{dt^2} + \frac{k_3}{m}\,\frac{dr}{dt} - \frac{k_2^2}{r^3} = 0 \quad .. \quad .. \quad (17)$$

which owing to the expression $1/r^3$ (which is due to the centrifugal force) cannot be solved in close form.

However, in the interval $(r_a - r_i)$ the centrifugal force

$$F = m\,\frac{k_2^2}{(r_v - \Delta r/2)^3} \quad .. \quad .. \quad (18)$$

can be considered as constant for each of the n strips of width Δr (see Fig. 7), so that for an arbitary interval v we get the differential equation

$$\frac{d^2r}{dt^2} + \frac{k_3}{m}\,\frac{dr}{dt} - \frac{F}{m} = 0 \; .. \quad .. \quad (19)$$

The solution is then, with the boundary conditions $t = 0$, $r = r_{v-1}$, and $w = w_{v-1}$:—

$$r = r_{\nu-1} + \frac{F}{k_3}\,t - \frac{F_m}{k_3^2}\,C_1\left(1 - e^{-\frac{k_3}{m}t}\right)$$
$$(\nu = 1, 2, \ldots n) \quad .. \quad .. \quad (20)$$

and the constant

$$C_1 = 1 - \frac{k_3}{F}\,w_{\nu\,1} \quad .. \quad .. \quad (21)$$

By neglecting d'Alembert's force $m\dfrac{d^2r}{dt^2}$ the solutions of eqs. (15) and (19) are still further simplified,

$$r = r_k\, e^{\frac{m\,k_1^2}{k_3}t} \qquad (0 < r < r_i) \quad .. \quad .. \quad (16a)$$

$$r = r_{\nu-1} + m\,\frac{k_2^2}{k_3}\,\frac{1}{(r_\nu - \Delta r/2)^3}\,t \qquad (r_i < r < r_a) \quad (20a)$$

From these two equations the time t_i can be calculated which the dust particle requires to reach r_i, and the time t_a it needs to pass from r_i to r_a

$$t_i = \frac{k_3}{m\,k_1^2}\,\log_e\frac{r_i}{r_k} \quad .. \quad .. \quad .. \quad (22)$$

For passing through the interval ν the particle requires the time

$$t_\nu = \frac{k_3}{k_2^2}\,\frac{\Delta r}{m}\,(r_\nu - \Delta r/2)^3$$

and thus, the time to reach r_a is

$$t_a = \frac{k_3}{k_2^2}\,\frac{\Delta r}{m}\sum_1^\nu (r_\nu - \Delta r/2)^3.$$

and for $\Delta r \to 0$ we can write

$$t_a = \frac{k_3}{k_2^2}\,\frac{r_a^4 - r_i^4}{4\,m} \quad .. \quad .. \quad (23)$$

Equations (16a) and (20a) refer to the radial movement only. In order to solve the problem of minimum bottom distance, a knowledge of the axial velocity in the separating chamber is required. It will suffice to obtain an approximate distribution of the axial velocities from limit values. When the core downward flow meets the upward flow through the bottom hole it is radially deflected by the latter. At this station the axial velocity must become zero, i.e. the velocity v_k at the bottom hole, which can be calculated from the pressure drop Δh_m, eq. (7) diminishes to zero. (Fig. 6). The velocity of the rising flow in the zone $(r_a - r_i)$ must, therefore, be accelerated from zero to the maximum value obtaining at the the outlet duct entrance.

$$c_{z\,\max} = \frac{Q}{(r_a^2 - r_i^2)\,\pi}$$

Strictly speaking, this is only valid for a stream filament at the centre (curve *a* in Fig. 6). Layers further outward might be decelerated by the core downward flow (curve *b*) or even accelerated by the rising flow directed towards the outlet duct (curve *c*). As the distribution of axial velocities between v_k and $c_{z\,\max}$ is unknown, we assume the most unfavourable case, namely that the particle travels through the chamber with the axial velocity v_k until it reaches r_i and with c_z until it reaches r_a. Then, the minimum bottom distance becomes

$$z_{\min} = v_k\,t_i + c_{z\,\max}\,t_a.$$

The first expression on the right hand side of eq. (24) represents the distance to the section where the two axial flows meet, the second term represents the distance from that section to the upper edge of the outlet duct.

Any dust particles smaller than $a_{\min}$ would pass through the outlet duct and remain in the clean gas.

A general relationship between grain size still capable of extraction and bottom distance can be obtained from eq. (24) in conjunction with eqs. (22), (23) and (14a), thus

$$z a^2 = 18\,\nu\,\frac{\gamma_1}{\gamma_s}\left[\frac{v_k}{k_1^2}\,\log_e\frac{r_i}{r_k} + \frac{c_{z\,\max}}{k_2^2}\,\frac{r_a^4 - r_i^4}{4}\right] \quad (25)$$

By means of this equation, disregarding the small variation of the velocities v_k and $c_{z\,\max}$ due to varying bottom distance, the collecting efficiency η_a can be calculated by equating the right hand side to a constant K.

A general expression for the collecting efficiency based on the probability curve [4] can be written as

$$\eta_a = 100\,e^{-x a^y} \quad .. \quad .. \quad .. \quad (26)$$

and with eq. (25)

$$\eta_a = 100\,e^{-x\,(K z)^{y/2}} \quad .. \quad .. \quad (27)$$

where x and y are exponents characteristic of the dust used.

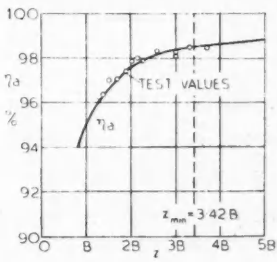

Fig. 8. **Collecting efficiency of cyclone as a function of the bottom distance.**

Eq. (27) has been plotted in Fig. 8 and test results (the same equipment as for Figs. 2 and 3 being used) plotted in the same figure are in closest agreement with calculated values. Other data in this case were :—

Weight of air per unit volume $\gamma_1 = 0{\cdot}0012$ gr./cu.cm
Weight of fly-ash per unit volume $\gamma_s = 1{\cdot}5$ gr./cu.cm.
Velocity head of cyclone $h_c = 10{,}000$ cm. air [column

$x = 0{\cdot}00357$.

$\gamma = 2{\cdot}08$ from Stokes's gravitational law.

Of practical importance is that part of the curve in Fig. 8 between $z = B$ and $z_{\min} = 3{\cdot}42\,B$.

It would be useless to exceed the value $z_{\min}$ because for a given cyclone with corresponding smallest grain size $a_{\min}$, optimum extraction is reached. On the contrary, the requirement for a high pressure drop Δh_a calls for a small value of the bottom distance z (see Chapter I), and a value somewhat smaller than $z_{\min}$ should be taken without, however, appreciably losing in efficiency.

It would be wrong to assume that for external suction the value of $z_{\min}$ calculated above could be reduced, since dust particles not yet removed from the chamber through the dust outlet slot are caught by the radial flow near the plate bottom, and thus they are driven into the rising core flow very much in the same way as particles re-entering the separating chamber through the bottom hole. (See Fig. 6).

III. The Reduction of Power Input.

Experience has shown that every attempt at reducing the power consumption of the cyclone by con-

structional means, such as guide vanes or diffusors in the separating chamber, reduced the effect of the rotational and centrifugal field respectively, and in so doing, lowered the collection efficiency. Therefore, power consumption could only be reduced by means external to the cyclone, and in no way reacting on internal flow conditions. It appears that to attain the desired effect, the only way is to transform the kinetic energy of the clean gas leaving the cyclone into pressure.

Although the clean gas which enters the outlet duct with the absolute velocity c_e loses some of its energy by friction along the duct, it is assumed that, on account of the small length of the duct, the gas leaving it still has the velocity c_e.

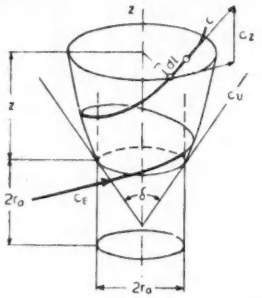

Fig. 9. Flow in extension of outlet duct.

If the cylindrical part of the outlet duct is extended by an arbitrarily shaped duct of increasing cross-section ($r = f(z)$, Fig. 9), the loss in the moment of momentum due to friction, neglecting internal friction and radial flows, can be calculated as follows :—

If at some station of the outlet duct the velocity is

$$c = \sqrt{c_u^2 + c_z^2} \quad .. \quad .. \quad (28)$$

the kinetic energy of the flow would be $H = c^2/2g$. Along the path dl this energy decreases by the amount

$$dH = -d\frac{c^2}{2g} \quad .. \quad .. \quad .. \quad (29)$$

consumed by friction, and which can also be written

$$dH = \lambda \frac{dl}{2r}\frac{c^2}{2g} \quad .. \quad .. \quad .. \quad (30)$$

where λ is the co-efficient of friction of the duct.

Eq. (29) and (30) can be combined, and on substituting for $dl = c/c_z\, dz$ and for $c_z = Q/r^2\pi$ we have

$$c = \frac{c_E}{1 + \frac{\lambda\pi}{4Q} c_E \int f(z)\, dz} \quad .. \quad .. \quad (31)$$

For a cylindrical extension, with $r = f(z) = \text{const.} = r_a$ and $Q = c_m r_a^2 \pi$ we get

$$c = \frac{c_E}{1 + \frac{\lambda c_E}{4 c_m r_a} z} \quad .. \quad .. \quad .. \quad (32)$$

The length of duct required to fully consume the moment of momentum (the so-called braking distance z_B) is obtained when the absolute velocity c is transformed into pure axial velocity c_m.

$$z_B = \frac{4 r_a}{\lambda}\left(1 - \frac{c_m}{c_z}\right) \quad .. \quad .. \quad (33)$$

Theoretically z_B should be infinite, as c and c_u respectively should decrease according to an exponential law. This is not the case, because a number of simplifying assumptions has been made. Test results actually show that z_B as calculated from eq. (33) is considerably on the high side because for practical purposes the moment of momentum has vanished when it ceases to be measurable.

Measurements of the pressure loss h_e taken on a cyclone with cylindrical outlet duct have shown that there is a certain duct length corresponding to minimum power expenditure. This diffusor effect is due to the fact that part of the peripheral velocity is transformed into pressure whilst the rest is consumed by friction. The best results correspond to a duct length of less than $4r_a$ beyond which value the gain obtainable is not commensurate with the expenditure in material and space.

The treatment of the problem of a conical extension of the outlet duct with an apex angle δ yields the following result :—

According to Fig. 9, $r = f(z) = r_a + z \tan \delta/2$. Whereas in the cylindrical duct the absolute velocity c tends towards a limiting value c_m, in the conical duct it tends towards zero. To allow of comparison with the cylindrical duct, it is assumed that the braking distance z_B is reached when c is transformed into c_m.

$$z_B = \left[\sqrt{\frac{8 \tan \delta/2}{\lambda}\left(1 - \frac{c_m}{c_E}\right) + 1} - 1\right] \operatorname{cotan} \delta/2 \quad .. \quad (34)$$

For $\delta = 0$ deg., the braking distance for a cylindrical duct can be obtained by eq. (33), and for $\delta = 180$ deg. we have $z_B = 0$ (Fig. 10). The most effective decrease of

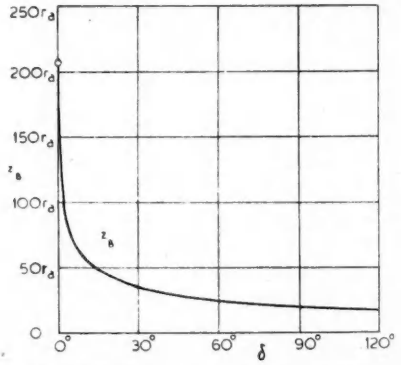

Fig. 10. Braking distance z_B as a function of the apex angle δ of outlet duct.

the braking distance is noticeable at small values of δ. This is also borne out by measurements of the pressure loss h_e where the greatest recuperation of pressure (abt. 20 per cent) is obtained for values of δ between 7 and 14 deg.

For $\delta = 0$ deg., i.e. in the cylindrical duct, about 10 per cent of the pressure is regained.

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A New Machine for the Dynamic Testing of Textiles and Rubber

By A. TÉNOT. (From *Le Génie Civil*, Vol. 124, No. 18, September 15, 1947, pp. 349-351, 4 illustrations.)

THE Établissements A. J. Amsler & Cie. in co-operation with the Établissements R. & H. Huber ("Pallas" tyres) have developed a machine for testing the toughness and the static and dynamic strength of rubber and fabric strips, and other materials used in tyre manufacture.

The machine is designed to enable measurements to be taken at normal temperatures and at the higher temperatures reached by tyres as a result of heat due to friction. The test pieces can be subjected to static and alternating loads with or without pre-stressing. The machine measures elastic and permanent deformations, and records the number of cycles to which the test-piece has been subjected.

The alternating load is supplied by a torsional exciter consisting of a vertical arm A (see Fig. 1) oscillating without friction about a horizontal hollow shaft B containing a torsionally elastic bar C attached at one end to the arm A and at the other to the frame D of the machine. The exciter vibrates owing to the unbalance of a rotating disk E which is situated on arm A at about one quarter of its length from the lower end. The unbalance of the disk can be adjusted by angular displacement of the two equal out-of-balance masses F and G on a circumference with the same centre as the disk. The disk is connected by a light, flexible belt drive I to a synchronous motor $\mathcal{J}$ rotating at 1,000 r.p.m.

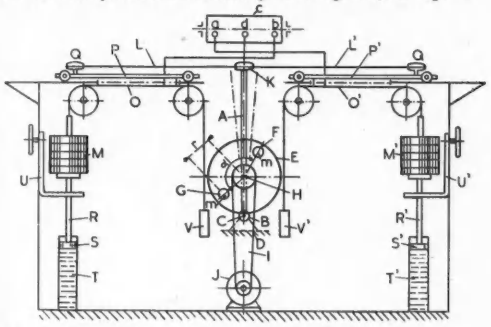

Fig. 1. **The Amsler machine for the dynamic testing of rubber and fabric strips.**

The test-piece, which is either 80 or 40 cm long, is attached to the upper end K of arm A. The two parts L and L' of the test-piece are subjected to an alternating load provided the static loads at their outer ends are at least equal to one half of the pulsating load supplied by the exciter. The static loads are provided by (1) the weights M and M' attached to the steel bands O and O'; (2) the carriages P and P' which are fixed on the steel bands and which can move horizontally along the frame of the machine, and (3) the attachments Q and Q'. The carriages move slightly during each load cycle, owing to the deformation of the test piece, but can be regarded as stationary in the vibrating system because of their comparatively great inertia.

To avoid shock effects during the rupture of one of the two parts of the test-piece, the weights M and M' are supported by rods R and R' provided with damping pistons S and S' moving within the oil-filled cylinders T and T', so that the weights can come to rest slowly on their supports U and U'. The system M, M', R, R' and S, S' is balanced by means of counterweights V and V'.

The masses M and M' consist of a number of cylindrical elements enabling the static load to be varied by 100 grams at a time up to a maximum load of 10 kg. The maximum (half total) amplitude of the deformation is 15 mm, and the pulsating force can be varied from 0·1 to 2·0 kg; the exciting frequency is 1,000 c.p.m., a 100-watt motor being used to actuate the system. Tests made with a cotton cord of type 23/5 under a static load of 3·8 kg and a superimposed alternating load of 2 kg (that is, with a resultant fluctuating load of 1·8 to 5·8 kg) produced rupture after 22,000 reversals.

When the system is vibrating without a test-piece, the inertia torque of the exciter is opposed by the shaft torque, owing to the stiffness of the torsional bar, and very little energy has to be supplied to keep it vibrating on its resonant frequency. The test-piece introduces an additional elastic force which alters the natural frequency of the system and which is compensated by the centrifugal force ϕ of the unbalanced disk : $\phi = mr\,\omega^2 \cos(\alpha/2)$, where m is the total mass of weight F and G, r is the radius of the circumference on which these weights are situated, ω is the angular velocity of the disk, and α is the angle between the two radii passing through the centres of F and G.

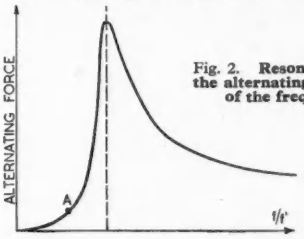

Fig. 2. **Resonance curve showing the alternating force as a function of the frequency ratio f/f'.**

The alternating force applied to the test-piece is determined by the horizontal component of ϕ and by the ratio $f/f' = \omega/\omega_n$ of the disk excitation frequency to the natural frequency of the torsional exciter with the test-piece; the alternating force varies therefore as shown in Fig. 2. To avoid the unstable conditions near the resonance peak, in the neighbourhood of which any accidental variation of the disk velocity can cause considerable changes in the dynamic force, the system is run on the lower part of the flank (point A in Fig. 2) of the resonance curve, between the origin and the steeper portion of the curve. The system is stable under these conditions and low disk velocities are then required. For stable functioning, the ratio of the elastic constants of the test-piece and the torsional bar should be as high as possible.

The alternating load can be varied by altering the angle between the two masses F and G (Fig. 1). The static or dynamic deformations of the two parts of the test-piece are recorded on paper by means of recording pens a and b mounted respectively on carriages P and P', the recording strip being on a drum which is driven by a synchronous motor through a gear train displacing the paper at speeds of 10 or 5 mm per minute, so that 1 mm displacement will correspond to either 1,000 or 2,000 reversals. The oscillating arm A also carries a stylus d which records the amplitude of the vibrations. The total number of reversals can be read off from the curves or recorded by a revolution counter with a synchronous motor. For tests at higher temperatures, each of the two parts of the test-piece is placed in a resistance-type electric oven, 20 cm in length : the temperature is controlled by a slider resistance and can reach 120 deg. C. The machine can also be used to determine the behaviour of bonded and vulcanized materials.

RUSSIA

Method for Computing Steam Power Plants Equipped with Air-cooled Condensers

By B. A. GRIGORYEV. (From *Izvestia Akademii Nauk*, No. 2, 1947, pp. 165-182, 5 illustrations.)

STEAM power plants equipped with condensers of the air-cooled, tubular or honeycomb type find wide-spread application in the field of condensing locomotives. Combined with the employment of medium and high initial steam pressures and temperatures it should be possible to use steam power plants with air-cooled condensers in automobiles, tractors, rail cars and other vehicles which are usually equipped with internal combustion engines.

The employment of steam propulsion plants for such purposes improves the traction characteristics of the power plant because of the resultant simplification of the means for power transmission, longer service life, and the possibility of using waste, or low grade, fuel.

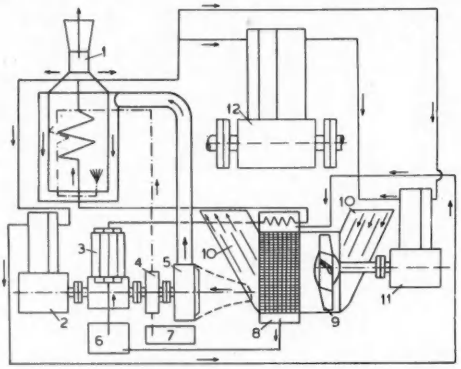

Fig. 1. **Diagram of high pressure steam prime mover plant with air condenser.**

(1) Steam generator; (2) Steam engine driving auxiliaries; (3) Feed pump; (4) Fuel pump; (5) Forced draught fan; (6) Hot-well; (7) Fuel tank; (8) Condenser; (9) Fan; (10) Air duct of condenser; (11) Engine driving condenser air fan; (12) Main prime mover.

The main items of a high pressure "power pack" with an air-cooled condenser (Fig. 1) comprise :

(a) the steam generator which is most frequently of the forced flow once through type ;
(b) the auxiliary machinery such as draught fans, high pressure feed-water pumps, fuel pumps, and stoker equipment ;
(c) the steam drive of the auxiliary machinery ;
(d) the plant consisting of the condenser, the fans, and the air inlet and outlet ducts ;
(e) the steam drive of the condenser fans ;
(f) the main steam propulsion unit.

Condenser Characteristics.

Condenser characteristics are expressed by the empirically established variation of the overall coefficient of heat transfer and of the frictional air flow resistance of the condenser. The latter is a function of the mass flow of the air based upon the frontal surface of the condenser, and of the type and depth of the condenser. It was found by other investigators that the coefficient of heat transfer from the steam to the tube wall is of little influence upon the overall coefficient so that the latter may be taken to equal the coefficient of heat transfer from the tube to the air. This coefficient can be expressed by the formula :

$$k \sim \alpha_2 = f(u_L, G) \quad .. \quad .. \quad (1)$$

where $u_L = v_L \gamma_L$ is the mass velocity of the air at the condenser inlet, based upon Nusselt's fundamental criterion:

$$Nu = f(Re, Pr, Gr, l/d \ldots) \text{ or } Nu = f(Re, Pr, l/d \ldots)$$

The coefficient of frictional air flow resistance can be expressed as

$$\zeta_K = \zeta_{0K} + \theta_K \quad .. \quad .. \quad (2)$$

where $$\theta_K = \frac{2(t_L' - t_L)}{T_L}$$

and $$\zeta_{0K} = f(u_L, G) \quad .. \quad .. \quad (4)$$

where ζ_{0K} is the air friction coefficient in the condenser. The air flow resistance therefore is

$$\Delta p_K = \zeta_K \frac{\rho_L v_L^2}{2} \quad .. \quad .. \quad (5)$$

where v_L is the air velocity in the condenser.

It is expedient to base the heat transfer coefficient and the temperature differential on the conditions prevailing at the frontal face of the condenser. With q denoting the coefficient of heat transfer F, in sq. m. denoting the frontal area of the condenser, t_s denoting the temperature of the condensing steam and t_L denoting the temperature of the atmospheric air at the condenser inlet and t_L' that at the condenser outlet,

$$Q = Fq(t_s - t_L) = Fq\Delta t \quad .. \quad (6)$$

where Q in cal/hr is the condenser duty.

The coefficient q can be obtained from the expression for the heat transfer in the condenser which reads :

$$Q = k\phi\, Hk \frac{(t_L' - t_L)}{\log_e \left(\frac{t_s - t_L}{t_s - t_L'} \right)} =$$

$$= k\phi\, S\, G\, F\, k \frac{\Delta t_L}{\log_e \left(\frac{1}{1 - \frac{\Delta t_L}{\Delta t}} \right)} \quad (7)$$

so that

$$Q = F v_L \gamma_L c_p \Delta t_L \times 3600 \quad .. \quad .. \quad (8)$$

It should be noted that the coefficient k_ϕ in eq. (7) is a fraction which is given by the ratio of the surface area actually swept by the atmospheric air to the total amount of air-exposed surface provided. By combining eqs. (6) and (8)

$$q = v_L \gamma_L c_p \frac{\Delta t_L}{\Delta t} 3600 \quad .. \quad .. \quad (9)$$

where $\Delta t_L/\Delta t$ represents the "coefficient of air utilization" the numerical value for which is obtained from eqs. (7) and (8) in the form of

$$\frac{\Delta t_L}{\Delta t} = 1 - \frac{10 \exp. v_L \gamma_L c_p \times 3600 \times 2{\cdot}303}{S k k_\phi G} \quad .. \quad (10)$$

The heat transfer capacity of the condenser can be computed as follows : For any steam plant cycle (Fig. 2) the following equation holds true :—

$$\frac{\Delta I D}{632} = \frac{N}{\eta_M} + \frac{N_{BY}}{\eta_{BY}} + \frac{N_B}{\eta_B} \quad .. \quad .. \quad (11)$$

where ΔI is the adiabatic heat drop in cal/kg., D is the amount of steam in kg. per hour, N is the useful effective output horse-power, η_M is the overall efficiency

of the main engine, N_{BY} is the power consumption of the air fan drive of the condenser with efficiency η_{BY}, and N_B is the power consumption of the steam drive for the auxiliary machinery with efficiency η_B.

The power requirements η_B of the auxiliary plant are assumed to be a function of the total steam consumption so that

$$N_B = RD \qquad .. \qquad .. \qquad (12)$$

Also

$$N_B = N_{BD} + N_H + N_G \qquad .. \qquad .. \qquad (13)$$

where N_{BD} is the power consumption of the draught plant, N_H is the power consumption of the feed-water pump, and N_G is the power consumption of the fuel pump.

Therefore

$$R = \frac{N_{BD} + N_H + N_G}{D} \text{(h.p. hr./kg.)} \qquad .. \qquad (14)$$

Furthermore

$$N_{BD} = \frac{W \alpha L_0 \Delta P_{BD}}{\eta_{BD} \gamma_L \times 75 \times 3600} = \frac{D \Delta i \alpha L_0 \Delta p_{BD}}{\eta_K Q_H{}^P \eta_{BD} \gamma_L \times 75 \times 3600} \qquad (15)$$

where W in kg./hr. is the fuel consumption, α is the excess air factor, L_0 is the theoretical amount of combustion air required in kg. of air per kg. of fuel, Δp_{BD} in kg./sq.m. is the total draught loss of the steam generator, η_K is the efficiency of the steam generator, $Q_H{}^P$ is the lower calorific value of the fuel in cal./kg., η_{BD} is the efficiency of the draught plant including reduction gear, γ_L is the density of the atmospheric air in kg./cu.m. Further

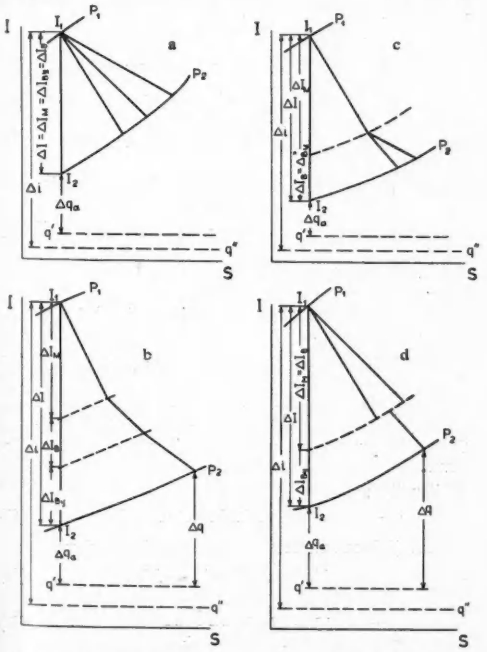

Fig. 2. **Possible cycle arrangements of steam power plant (Fig. 1) with air-cooled condenser.**
I = Heat content. S = Entropy.
(a) Parallel cycle; (b) Series cycle; (c) and (d) Mixed cycles.
Suffixes: M refers to main engine.
BY " " fan drive.
H " " auxiliary drive.

$$N_H = \frac{k_H p_1 D \times 10^4}{\gamma_B \eta_H \times 75 \times 3600} \qquad .. \qquad (16)$$

where γ_B is the unit weight of the feed-water in kg./cu.m. and where $k_H = \frac{p_H}{p_1}$ is the ratio of the pressure developed by the feed pump and the pressure at the steam generator feed-water inlet; and η_H is the efficiency of the feed pump including reduction gear. The power consumption of the fuel oil pump is given by

$$N_G = \frac{W p_G}{\eta_G \gamma_G \times 75 \times 3600} = \frac{D \Delta i p_G}{\eta_K Q_H{}^P \gamma_G \eta_G \times 75 \times 3600} \qquad (17)$$

where p_G is the pressure developed by the fuel pump, η_K is the boiler efficiency, γ_G is the specific weight of the fuel oil in kg./cu.m., and η_G is the efficiency of the fuel pump including reduction gear.

By substituting the respective expressions for N_{BD}, N_H and N_G in eq. (14) the latter can be written

$$R = a \Delta i + b p_1 \qquad .. \qquad .. \qquad (18)$$

where

$$a = \left(\frac{\alpha L_0 \Delta p_{BD}}{\gamma_L \eta_{BD}} + \frac{p_G}{\gamma_G \eta_G} \right) \frac{1}{\eta_K Q_H{}^P \times 75 \times 3600} \qquad (19)$$

and $$b = \frac{k_H \times 10^4}{\gamma_B \eta_H \times 75 \times 3600} \qquad .. \qquad .. \qquad .. \qquad (20)$$

By combining eqs. (11) and (12)

$$D = \frac{\frac{N}{\eta_M} + \frac{N_{BY}}{\eta_{BY}}}{\frac{\Delta I}{632} - \frac{R}{\eta_B}} \qquad .. \qquad .. \qquad .. \qquad (21)$$

The required heat transmission capacity of the condenser can be expressed by

$$Q = D \Delta q \qquad .. \qquad .. \qquad .. \qquad (22)$$

where Δq is defined by the cycle conditions as indicated in Fig. 2.

The author goes on to investigate in detail the four basic cycle arrangements as outlined in the *I-S* diagrams reproduced in Fig. 2. In the cycle Fig. 2a both main engine and auxiliary steam drives are supplied with live steam of pressure p_1 and they exhaust at pressure p_2. In Fig. 2b live steam of pressure p_1 is partially expanded in the main engine from which it is supplied as back pressure to the auxiliary steam drives from whence it passes as back pressure to the steam drive for the condenser fan. The exhaust from the latter is then rejected to the condenser. In Fig. 2c the back pressure steam from the main engine enters the steam drives of the condenser fan and of the other auxiliaries at the same pressure; while in Fig. 2d the high pressure live steam is expanded to the same back pressure in the main engine as well as in the steam drives of the auxiliaries, whilst the steam drive of the condenser fan is operated by the back pressure steam.

Parallel Operation of Main engine and Auxiliaries including Fan Drive (Fig. 2a).

It can be written that

$$\frac{\Delta I D \eta_{ae}{}^{0/}}{632} = \frac{\Delta I D_M \eta_M{}^{0/}}{632} + \frac{\Delta I D_{BY} \eta_{BY}{}^{0/}}{632} + \frac{\Delta I D_B \eta_B{}^{0/}}{632} \qquad (23)$$

where η_{av}^{0f} is the average efficiency ratio and η_M^{0f} that of the main engine, η_{BY}^{0f} that of the auxiliary drive and η_B^{0f} that of the fan drive.

Hence

$$\eta_{av}^{0f} = \frac{1}{D}(D_M \eta_M^{0f} + D_{BY} \eta_{BY}^{0f} + D_B \eta_B^{0f}) \quad (24)$$

On the other hand

$$\Delta q = I_1 - \Delta I \eta_{av}^{0f} - q' = \Delta I (1 - \eta_{av}^{0f}) + \Delta q_a \quad \text{(See Fig. 2)} \quad .. \quad (25)$$

The hourly steam consumption of the individual engines is

$$D_M = \frac{N \times 632}{\Delta I \eta_M} \quad .. \quad .. \quad (26)$$

$$D_{BY} = \frac{N_{BY} \times 632}{\Delta I \eta_{BY}} \quad .. \quad .. \quad (27)$$

$$D_B = \frac{R D \times 632}{\Delta I \eta_B} \quad .. \quad .. \quad (28)$$

Equation (25) can therefore be written :

$$\Delta q = \left(\Delta I + \Delta q_a - R \times 632 \frac{\eta_B^{0f}}{\eta_B}\right) - \frac{632}{D}\left(N \frac{\eta_M^{0f}}{\eta_M} + N_{BY} \frac{\eta_{BY}^{0f}}{\eta_{BY}}\right) \quad (29)$$

Series Operation of Main Engine and Auxiliaries.

It will be assumed that within the working range of the *I-S* diagram the isobars corresponding to inlet and condenser pressure respectively are parallel with each other. This assumption leads to some slight error, the computed values of the adiabatic heat drop being slightly smaller than the actual values. Thus in the case of initial steam conditions ranging from 70 to 120 atmospheres and 400 to 500 deg. C., and of final steam conditions of 1 to 3 atmospheres at the condenser, the error will be in the neighbourhood of one per cent or less. The error, however, increases with lower initial pressures, and for the steam conditions of the FD-k condensing locomotive (initial steam conditions : 16 atmospheres, 350 deg. C. ; final steam condition : 1·1 atmospheres) the error amounts to 2·5 per cent which causes in turn an error of 1·5 per cent in the final result. Where greater accuracy is required, this error can be minimized by introducing a mean correction factor.

From Fig. 2b it is obvious that

$$\Delta q = \Delta I - \Delta I_M \eta_M^{0f} - \Delta I_B \eta_B^{0f} - \Delta I_{BY} \eta_{BY}^{0f} + \Delta q_a \quad .. \quad .. \quad .. \quad (30)$$

and the corresponding adiabatic heat drops are

$$\Delta I_M = \frac{N \times 632}{D \eta_M} \quad .. \quad .. \quad (31)$$

$$\Delta I_B = \frac{R \times 632}{\eta_B} \quad .. \quad .. \quad (32)$$

$$\Delta I_{BY} = \frac{N_Y \times 632}{D \eta_{BY}} \quad .. \quad .. \quad (33)$$

These equations prove that eq. (29) is valid for this case also, and the same is true for the mixed cycles outlined in Fig. 2c and 2d. Consequently the condenser duty with all three different cycle arrangements will be given by

$$Q = \left(\frac{1}{\frac{\Delta I}{632} - \frac{R}{\eta_B}}\right)\left\{\frac{N_{BY}}{\eta_{BY}}\left[\Delta q_{BY} + R\frac{632}{\eta_B}(\eta_{BY}^{0f} - \eta_B^{0f})\right] + \frac{N}{\eta_M}\left[\Delta q_M + R\frac{632}{\eta_B}(\eta_M^{0f} - \eta_B^{0f})\right]\right\} \quad (34)$$

where

$$\Delta q_{BY} = \Delta I (1 - \eta_{BY}^{0\cdot}) + \Delta q_a \quad .. \quad .. \quad (35)$$

and

$$\Delta q_M = \Delta I (1 - \eta_M^{0f}) + \Delta q_a \quad .. \quad .. \quad (36)$$

Power Consumption of the Condenser Fans.

The power consumption of the condenser fan is given by

$$N_{BY} = \frac{\eta_{BY}}{\eta_M} \frac{\left[\Delta q_M + R\frac{632}{\eta_B}(\eta_M^{0f} - \eta_B^{0f})\right]}{\left[\Delta q_{BY} + R\frac{632}{\eta_B}(\eta_{BY}^{0f} - \eta_B^{0f})\right]} \times \left\{Fq\left[\frac{\Delta t}{\Delta q_M + R\frac{632}{\eta_B}(\eta_M^{0f} - \eta_B^{0f})}\right] \times \left(\frac{\Delta i \eta_M}{632} - R\frac{\eta_M}{\eta_B}\right) - N\right\} \quad (37)$$

Where η_{BY} is the efficiency of the fan drive and η_M is the efficiency of the main steam power unit. For the sake of simplicity the following coefficients will be introduced, to wit :

$$k_0 = \frac{\eta_{BY}}{\eta_M} \frac{\left[\Delta q_M + R\frac{632}{\eta_B}(\eta_M^{0f} - \eta_B^{0f})\right]}{\left[\Delta q_{BY} + R\frac{632}{\eta_B}(\eta_{BY}^{0f} - \eta_B^{0f})\right]} \quad (38)$$

which expresses a ratio and is therefore dimensionless,

$$k_k = \frac{\Delta t}{\Delta q_M + R\frac{632}{\eta_B}(\eta_M^{0f} - \eta_B^{0f})} \quad \text{[deg. C., kg./cal]} \quad (39)$$

can be termed the " coefficient of condensation,"

$$k_M = \frac{\Delta I \eta_M}{632} - \frac{\eta_M}{\eta_B} R \quad \text{[h.p. hr./kg.]} \quad (40)$$

which is the output coefficient characterizing the efficiency of steam utilization for power generation in the main propulsion plant and in the condenser fan drive.

Finally :

$$k_p = k_K k_M \quad \text{[deg. C., h.p., hr./cal]} \quad .. \quad (41)$$

Eq. (37) can therefore be written in the form

$$N_{BY} = k_0 (F q k_p - N) \quad .. \quad .. \quad (42)$$

or

$$N = F q k_p - \frac{N_{BY}}{k_0} \quad .. \quad .. \quad (43)$$

Eq. (42) can also be written in the dimensionless form

$$\frac{N_{BY}}{N} = k_0 \left(\frac{F q k_p}{N} - 1\right)$$

and by introducing

$$\frac{N_{BY}}{N} = \bar{M} \text{ and } \frac{F q k_p}{N} = \bar{K}, \bar{M} = k_0(\bar{K} - 1) \quad .. \quad (44)$$

In view of the fact that the influence of the condenser upon the resistance of the air duct system is difficult to assess, it will be found more convenient to determine experimentally the air resistance coefficient for the condenser installation as a whole. In this way the air resistance coefficient of the air duct can be obtained in conjunction with condensers of different types and dimensions. In order to establish an analytical expression, all air resistance co-efficients will be based upon the air velocity prevailing at the front end of the condenser. On the basis of formulae (3) and (4) it can be written :

$$\zeta = (\xi_{0BP} + \zeta_{0K}) + (\theta_{BP} + \theta_K) = \xi_0 + \theta \quad .. \quad (45)$$

where, neglecting any non-isothermal flow of the air towards the exit,

$$\theta = \frac{\Delta t_1}{T_1}(2 + \zeta_{OUT}) \quad .. \quad .. \quad (46)$$

$$\zeta_{0BP} = \zeta_{IN} + \zeta_{OUT} + \Sigma \zeta_i \quad .. \quad (47)$$

Where ζ_{0BP} is the overall air resistance coefficient of the air duct ζ_i is the internal resistance coefficient.

The difference $t_L' - t_L = \Delta t_L$ is obtained from eq. (10). In view of the high Reynolds numbers involved in the case considered, it will be safe to assume the air resistance coefficient remains unaffected by the air velocity. The air flow resistance can therefore be expressed by

$$\Delta p = \zeta \frac{\rho_L v_L^2}{2} \quad .. \quad .. \quad (48)$$

where ρ_L is the density of the air in kg. sec^2/m^4.

The power requirements of a pressure blower forcing the air through the condenser will thus be

$$N_{BY} = \frac{\Delta p\, Q_L}{75\, \eta_{BP}} = \frac{\rho_L v_L^3 F \zeta}{2 \times 75 \times \eta_{BP}} \quad .. \quad (49)$$

where η_{BP} is the efficiency of blower and reduction gear, and the power required by a suction fan drawing the air through the condenser will be

$$N_{BY} = \frac{\rho_L v_L^3 F \zeta}{2 \times 75 \times \eta_{BP}} \left(1 + \frac{\Delta t_L}{T_L}\right) \quad .. \quad (50)$$

The efficiency of the main propulsion plant is given by

$$\eta = \frac{632\, N}{W\, Q_H{}^P} = \frac{632\, N \eta_K}{D\, \Delta i} \quad .. \quad .. \quad .. \quad (51)$$

where η_K is the boiler efficiency.

By introducing into this equation the expression for D given by eq. (21) and the coefficient k_M from eq. (40) and putting $\eta_{BY}/\eta_M = \lambda$

$$\eta = \frac{632\, \eta_K k_M}{\Delta i} \left(\frac{1}{1 + N_{BY}/N\lambda}\right) .. \quad .. \quad (52)$$

This equation shows η to be the product of two dimensionless factors, the first of which is

$$\eta_{LIMIT} = \frac{632\, \eta_K k_M}{\Delta i} \quad .. \quad .. \quad (53)$$

This factor is an exclusive function of the parameters of the plant and of the steam cycle employed. Moreover, it represents the efficiency of the plant if no power were expended upon the operation of the condenser. The second factor

$$\bar{H} = \frac{1}{1 + N_{BY}/N\lambda} \quad .. \quad .. \quad (54)$$

characterizes the power expended on operation of the condenser. Eq. (52) can also be written in the form

$$\eta = \eta_{LIMIT}\, \bar{H} \quad .. \quad .. \quad (55)$$

RUSSIA

Automatic Control of Hydro-electric Installations

By V. V. HIRSCHBERG. (From *Vestnik Electro-Promyshlennosti*, No. 5, 1947, pp. 13-18, 6 illustrations.)

ALTHOUGH an automatic hydro-electric station was first constructed as early as 1898, and such stations became common, there was, until recently, a tendency to confine the fully automatic design mainly to small stations where the resulting saving in attendance costs is particularly important. By the adoption of automatic control the total cost per unit may be reduced by 80 per cent.

But now " automatization " has been applied to the largest and most important installations, including the Boulder Dam and Grand Coulee. It reduces the time required to put a unit into service, renders it more reliable, improves load distribution and overall efficiency, and reduces attendance costs.

TYPES OF AUTOMATIC CONTROL.

The various types of automatic control in vogue can be classified under three headings as follows :—

1. *The " sequence " system* in which the various mechanisms to be operated to bring the set into service are separately actuated and put into operation one after another, and in which the commencement of each operation must wait until the previous operation is completed.

Lower Waikazemoana (Metropolitan Vickers) is an example of this system, which is slow in putting the set into service ; this is a drawback for very large and important units.

2. *The " controller " system* in which a slowly moving controller sets in motion various groups of isolated or interdependent operations at each successive position.

This system was much favoured by the A.E.G. and the diagram (Fig. 1) for the Shinonogawa hydro-electric plant is typical. It has the advantage that it avoids the use of interlocks and time relays, since the controller itself is a timing and interlocking mechanism, and of subdividing the whole process into a number of simple sequences. The position of the controller serves to indicate how far the starting process has advanced so that, if anything goes wrong, the mechanism at fault is easily traced. However, the controller, while simplifying the electrical system, is quite a complicated mechanism.

3. *The " parallel " system*, in which as many control mechanisms as possible are set in motion at the same time and the " sequence " principle is reduced to the minimum.

This is particularly favoured for large installations in Russia now, and the accompanying diagram (Fig. 2) for the Rybinsk plant illustrates its operating principle. It combines the advantages of the first two systems but eliminates their drawbacks.

The readiness of the unit for starting is signalled by a single " Permission to start " relay and there is a selector switch with a signal lamp which allows for the quick discovery of faults if " permission to start " is not available.

CONTROLLING MECHANISMS.

The control of automatic hydro-electric sets is effected by four regulating devices :—

(a) Turbine speed governors ;
(b) Power and frequency regulators ;
(c) Excitation control mechanisms ;
(d) Robot operators.

Their main lines of development are as follows :—

Speed governors are at present mainly hydro-mechanical, very complicated and insufficiently quick-acting to permit close frequency control without the use of a special apparatus. There is scope for a better incorporation of electrical devices, both to render them more adaptable to automatic controls and to make them simpler and more sensitive.

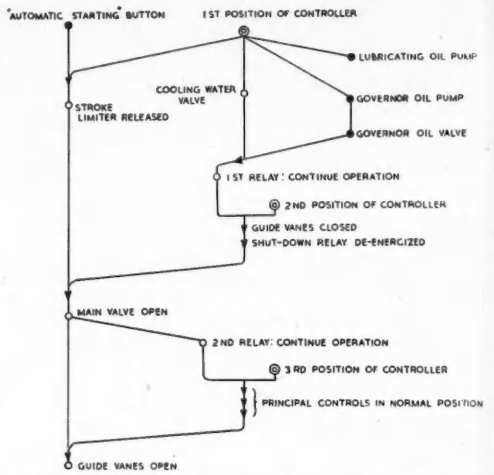

Fig. 1. First three stages of five-stage automatic controller starting of Shinonagawa hydro-electric plant.

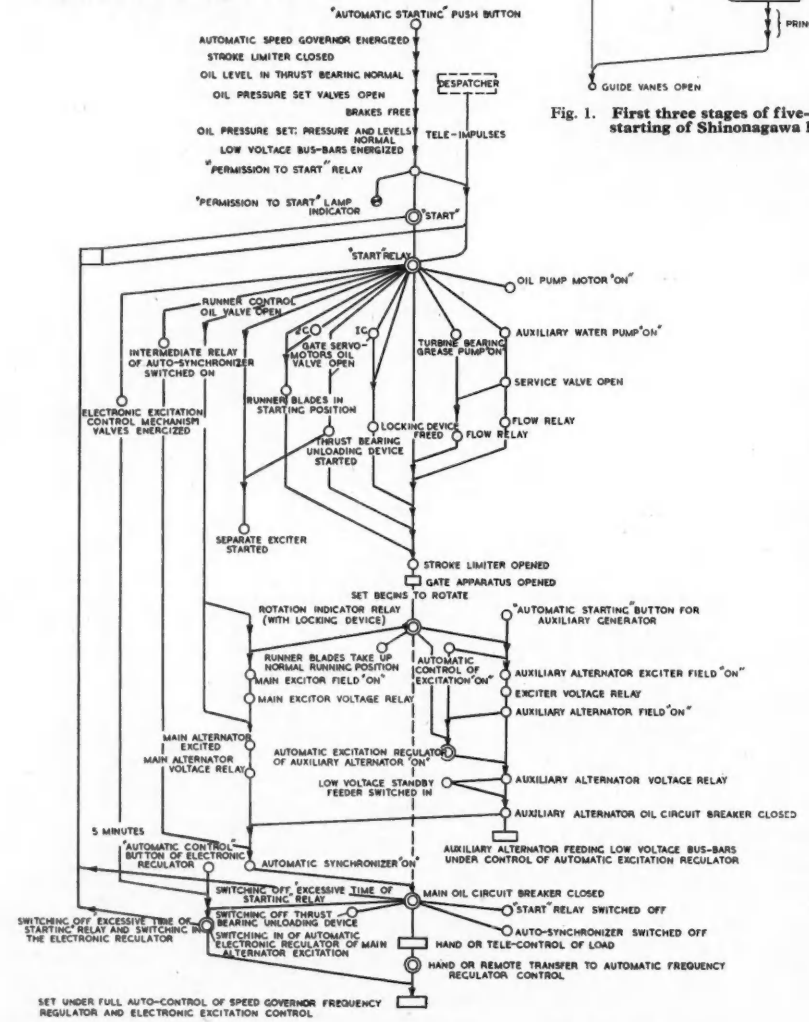
"AUTOMATIC STARTING" PUSH BUTTON
AUTOMATIC SPEED GOVERNOR ENERGIZED
STROKE LIMITER CLOSED
OIL LEVEL IN THRUST BEARING NORMAL
OIL PRESSURE SET VALVES OPEN
BRAKES FREE
OIL PRESSURE SET; PRESSURE AND LEVELS NORMAL
LOW VOLTAGE BUS-BARS ENERGIZED
"PERMISSION TO START" RELAY
DESPATCHER
TELE-IMPULSES
"PERMISSION TO START" LAMP INDICATOR
"START"
"START" RELAY
OIL PUMP MOTOR "ON"
RUNNER CONTROL OIL VALVE OPEN
AUXILIARY WATER PUMP "ON"
INTERMEDIATE RELAY OF AUTO-SYNCHRONIZER SWITCHED ON
GATE SERVO-MOTORS OIL VALVE OPEN
TURBINE BEARING GREASE PUMP "ON"
SERVICE VALVE OPEN
ELECTRONIC EXCITATION CONTROL MECHANISM VALVES ENERGIZED
RUNNER BLADES IN STARTING POSITION
LOCKING DEVICE FREED
FLOW RELAY
THRUST BEARING UNLOADING DEVICE STARTED
SEPARATE EXCITER STARTED
STROKE LIMITER OPENED
GATE APPARATUS OPENED
SET BEGINS TO ROTATE
ROTATION INDICATOR RELAY (WITH LOCKING DEVICE)
"AUTOMATIC STARTING" BUTTON FOR AUXILIARY GENERATOR
RUNNER BLADES TAKE UP NORMAL RUNNING POSITION
AUTOMATIC CONTROL OF EXCITATION "ON"
MAIN EXCITOR FIELD "ON"
AUXILIARY ALTERNATOR EXCITER FIELD "ON"
MAIN EXCITOR VOLTAGE RELAY
EXCITER VOLTAGE RELAY
AUXILIARY ALTERNATOR FIELD "ON"
MAIN ALTERNATOR EXCITED
MAIN ALTERNATOR VOLTAGE RELAY
AUTOMATIC EXCITATION REGULATOR OF AUXILIARY ALTERNATOR "ON"
LOW VOLTAGE STANDBY FEEDER SWITCHED IN
AUXILIARY ALTERNATOR VOLTAGE RELAY
AUXILIARY ALTERNATOR OIL CIRCUIT BREAKER CLOSED
5 MINUTES
"AUTOMATIC CONTROL" BUTTON OF ELECTRONIC REGULATOR
AUTOMATIC SYNCHRONIZER "ON"
AUXILIARY ALTERNATOR FEEDING LOW VOLTAGE BUS-BARS UNDER CONTROL OF AUTOMATIC EXCITATION REGULATOR
SWITCHING OFF EXCESSIVE TIME OF STARTING RELAY
MAIN OIL CIRCUIT BREAKER CLOSED
SWITCHING OFF THRUST BEARING UNLOADING DEVICE
"START" RELAY SWITCHED OFF
SWITCHING OFF EXCESSIVE TIME OF STARTING RELAY AND SWITCHING IN THE ELECTRONIC REGULATOR
SWITCHING IN OF AUTOMATIC ELECTRONIC REGULATOR OF MAIN ALTERNATOR EXCITATION
AUTO-SYNCHRONIZER SWITCHED OFF
HAND OR TELE-CONTROL OF LOAD
HAND OR REMOTE TRANSFER TO AUTOMATIC FREQUENCY REGULATOR CONTROL
SET UNDER FULL AUTO-CONTROL OF SPEED GOVERNOR FREQUENCY REGULATOR AND ELECTRONIC EXCITATION CONTROL

Fig. 2. Sequence of operations on automatic starting of the Rybinsk hydro-electric plant.

The latest excitation control mechanisms are electronic, but have a number of defects, such as the necessity for " warming up," the instability of the thyratrons, excessive complication, and the need for numerous protective devices.

In addition to speed and voltage control it is necessary to have regulating devices to adjust the division of load and power factor between sets, sometimes taking into account the turbine load-efficiency curve. Such regulators put the finishing touch to a system of automatic control and greatly reduce the demand for highly skilled personnel.

The robot-operator is a recent idea, designed to eliminate completely the human operator and to adjust automatically the running of the system in accordance with a set scheme to give the best conditions of operation taking into account such factors as low and high water, the efficiency of individual units, the desire to run certain units for peak load only, and so on.

A Calculation of Internal Stresses Due to Cold Extension or Compression

By H. BRANDENBERGER. (From *Schweizer Archiv*, Vol. 13, Nos. 8 and 9, August and September, 1947, pp. 232-238, and 268-275, 12 illustrations.)

ELONGATION beyond the elastic limit permanently deforms a tensile test piece, which, when unloaded, contains equally distributed internal compression stresses. These internal stresses increase the yield strength in tension but reduce the compressive yield strength below the value attained when the material is not pre-stressed. The general theory of elasticity equates the external unit load to the inner stress, and the stresses are presumed to be zero when the loading is not applied. However, precipitation or grain growth may tend to increase the lattice volume and be resisted by the rigid matrix structure so that elastic internal "volume" stresses are set up. Cold work may deform the shape of the matrix but only elastically strain the lattice volume so that an internal stress is set up between the deformed matrix and the lattice which tends to return to its original size when the load is taken off.

I. VOLUME AND FORM STRESSES.

In the following calculations[1,2] no attempt is made to explain the micro-mechanism of plastic deformation ; its existence is taken for granted and only experimentally measurable changes are considered. Assuming uni-directional elastic tension, the principal strains e and stresses σ are obtained from

$$e_1 = \frac{\sigma_1}{E} \; ; \; e_2 = e_3 = -\frac{e_1}{m} = -\frac{\sigma_1}{mE} \; ; \; \sigma_2 = \sigma_3 = 0 \quad (1)$$

and the volume change ΔV of a unit particle from

$$\Delta V = e_1 + e_2 + e_3 = \frac{\sigma_1}{E} \times \frac{m-2}{m} \neq 0 \text{ since } m \neq 2 \quad (2)$$

where m = Poisson's ratio = 10/3 for steel, and E = Young's modulus = 30×10^6 lb. per sq. in. for steel. The strain system of the unit particle can therefore also be obtained by superimposing a "volume" strain system, which produces a change in volume only, and a "form" strain system which deforms the shape of the particle without a change of volume. Stress systems can be co-ordinated to these part strain systems. Homogeneous deformations only (and not bending or torsion) are considered and an even distribution of stresses can be assumed. Any existing inhomogeneities will be irregular and stresses on any measurable volumes will therefore be identical. Only principal directions of stress of the external loading system are dealt with in the calculations. Shear stresses are thus left out because those shear stresses which completely determine a given state of stress are not necessarily the maximum shear stresses occurring in the stressed material[1].

If absolute volume elasticity is assumed, i.e. if the unstressed volume is always constant, then any volume stress system producing a certain volume change ΔV (eq. 2) can always be substituted by another system producing the same volume change and containing three equal mean stress values in the three principal direction

$$p_1 = p_2 = p_3 = p = \frac{\sigma_1 + \sigma_2 + \sigma_3}{3} \quad .. \quad (3)$$

This "volume" stress system superimposed on a "form" stress system determines the complete state of strain produced by the outer loading, i.e. the change of size and shape of the unit particle. With σ_1 σ_2 σ_3, as the known principal stresses due to external loading, and p as the mean volume stress, the "form" stresses become

$$\sigma_1' = \sigma_1 - p \; ; \; \sigma_2' = \sigma_2 - p \; ; \; \sigma_3' = \sigma_3 - p \; ;$$
$$\sigma_1' + \sigma_2' + \sigma_3' = 0 \; .. \; (4)$$

and the deformations

$$e_1 = \frac{p}{E''} + \frac{\sigma_1 - p}{E'} \; ; \quad e_2 = \frac{p}{E''} + \frac{\sigma_2 - p}{E'} \; ;$$
$$e_3 = \frac{p}{E''} + \frac{\sigma_3 - p}{E'} \; ; \; e_1 + e_2 + e_3 = \Delta V \quad (5)$$

where E' = the modulus of elasticity for form stresses, E'' = the modulus of elasticity for volume stresses. Combining eqs. (1) and (5) for uni-directional tension (Fig. 1), we obtain

$$e_1 = \frac{\sigma_1}{3E''} + \frac{2\sigma_1}{3E'} = \frac{\sigma_1}{E} \; ;$$
$$e_2 = e_3 = \frac{\sigma_1}{3E''} - \frac{\sigma_1}{3E'} = -\frac{\sigma_1}{mE} \quad .. \quad (6)$$

$$E' = E\,\frac{m}{m+1} \text{ and } E'' = E\,\frac{m}{m-2} \quad .. \quad (7)$$

For steel, $E' = \frac{10}{13}$, $E = 23 \times 10^6$ lb. per sq. in.,

$$\text{and } E'' = \frac{10}{4} E = 75 \times 10^6 \text{ lb. per sq. in.} \quad (8)$$

If e_1, e_2, e_3 are known,

$$p = \frac{\Delta V}{3} E'' = \frac{e_1 + e_2 + e_3}{3} E'' = e\,E'' \quad .. \quad (9)$$

the form strains $e_1' = e_1 - e$, $e_2' = e_2 - e$, $e_3' = e_3 - e$, the form stresses $\sigma_1' = e_1' E'$, $\sigma_2' = e_2' E'$; $\sigma_3' = e_3' E'$, and the principal stresses from eq. (4).

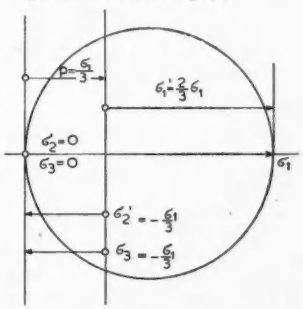

Fig. 1. Representation of volume and form stresses by Mohr's stress circle for simple tension in direction 1. (See also eqs. 1, 3, and 4.)

σ_1, σ_2, σ_3 = principal stresses due to external loading,
$p = p_1 = p_2 = p_3$ = mean volume stress,
σ_1', σ_2', σ_3' = form stresses.

Therefore even uni-directional loads always produce a three-dimensional state of stress. Only where the principal stresses lead to pure volume changes ($\sigma_1 = \sigma_2 = \sigma_3 = p$) or pure form changes ($\sigma_1 + \sigma_2 + \sigma_3 = 0$) can the deformations of the unit particle be directly calculated by means of a single defined modulus of elasticity. Otherwise, the strain in any principal stress direction is the sum of the form strain in that direction

and the mean volume strain (eq. 5). For a single external load, the volume and form strains add up in the load direction, and the volume and form stresses in any other principal direction are equal, yet, because of the difference in elasticity coefficients, a strain exists in that direction, eq. (6).

II. LATTICE AND SPACE STRESSES.

Complete stress and strain systems can be obtained by another superposition. Again, no metallurgical considerations are involved. Assuming that particles arranged on the faces of a unit cube can oscillate elastically about their equilibrium position and that their elastic deflection produces an elastic "lattice" stress, $\sigma_g = e E_g$, then the deflection also produces a "space" stress σ_R, i.e. a reaction of the whole matrix against any volume change of the unit cube. Lattice and space stresses in each of the three principal directions completely determine the behaviour of the material under stress. Space stresses σ_R are not identical with volume stresses p because these may produce lattice stresses σ_g as well as space reactions σ_R. The principal strains e_1, e_2, e_3 are produced by lattice stresses only, whilst the space stresses react only against overall volume change and are equal for the three principal directions

$$\sigma_{R1} = \sigma_{R2} = \sigma_{R3} = \sigma_R = \frac{\Delta V}{3} E_R = e E_R \quad .. \quad (10)$$

For three-dimensional tension :

$$\sigma_1 = \sigma_2 = \sigma_3 = p \; ; \; \sigma_1' = \sigma_2' = \sigma_3' = 0 \; ;$$
$$e_1 = e_2 = e_3 = p/E'' \; ; \; \sigma_{g1} = \sigma_{g2} = \sigma_{g3} = e E_g \; ;$$
$$\sigma_R = p - \sigma_g = eE'' - eE_g = eE_R ; \; E'' = E_R + E_G.$$

For simple, uni-directional tension : (Fig. 2)

$$e_1 = \frac{\sigma_1}{E} = \frac{\sigma_{g1}}{E_g} \; ; \; e_2 = e_3 = -\frac{e_1}{m} = -\frac{\sigma_1}{mE} =$$
$$= \frac{\sigma_{g2}}{E_g} = \frac{\sigma_{g3}}{E_g} \quad .. \quad .. \quad (11)$$

$$\sigma_R = \sigma_1 - \sigma_{g1} = -\sigma_{g2} = -\sigma_{g3} = \frac{e_1 + e_2 + e_3}{3} E_R =$$
$$= \frac{e_1}{3}\left(1 - \frac{2}{m}\right) E_R \quad .. \quad (12)$$

Combining eqs. (11) and (12)

$$E_g = E\frac{m}{m+1} = E' \; ; \; E_R = E\frac{3m}{(m+1)(m-2)} \; .. \quad (13)$$

and for steel,

$$E_g = E' = 23 \times 10^6 \text{ lb. per sq. in,}$$
$$E_R = \frac{45}{26} E = 52 \times 10^6 \text{ lb. per sq. in. .. } (14)$$

E_g equals E' since both characterize the resistance of a unit cube face against change of position. If the external load produces no change of volume, then $E_g = E'$ characterizes the resistance of the material against the effect of this load. If the load produces a volume change, then lattice stresses and a space reaction are generated and the mean volume stress is

$$p_1 = p_2 = p_3 = p = e E'' = e E_g + e E_R = \sigma_g + \sigma_R \; (15)$$

However, even without external loading, internal stresses may exist when space reactions oppose lattice position changes which include a volume change, as e.g. after cold deformation.

$$\sigma_R = e E_R = -\sigma_g \quad .. \quad .. \quad (16)$$

Unless the material is annealed, this internal state of stress must affect the behaviour of the material under load.

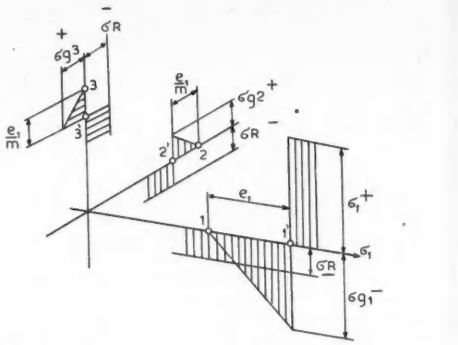

Fig. 2. **Axonometric representation of space stress σ_R, lattice stresses σ_{g1}, σ_{g2}, σ_{g3}, and of the particles 1′, 2′, 3′, situated on the end faces of a unit cube under uni-directional tension σ_1. 1, 2, 3 = unstressed positions of the particles 1′, 2′, 3′.**

III. STRESSES IN THE YIELD RANGE.

The unit cube, position 1 2 3 in the unstressed state (Fig. 2), is stretched by uni-directional loading σ_1 into position 1′ 2′ 3′ with $11' = e_1$, $22' = 33' = -e_1/m$. The positional change of the cube faces produces lattice stresses proportional to the change, and the change of cube volume sets up a space stress acting equally in the three principal directions. Equilibrium in position 1′ 2′ 3′ requires

$$\sigma_1 + \sigma_{g1} + \sigma_R = 0 \; ; \; \sigma_{g2} + \sigma_R = 0 \; ; \; \sigma_{g3} + \sigma_R = 0 \quad (17)$$

$$\sigma_{g1} = -e_1 E_g \; ; \; \sigma_{g2} = \sigma_{g3} = +\frac{e_1}{m} E_g = \frac{-\sigma_{g1}}{m} = \frac{e_1 E}{m+1} \; ;$$
$$\sigma_R = -\sigma_{g2} \quad .. \quad (18)$$

On yielding, the unstressed position changes. Yielding occurs first in direction 1 ($\sigma_{g1} > \sigma_{g2}$) and the end face position for $\sigma_{g1} = 0$ moves to 1* (Fig. 3). Simultaneously, the volume of the unit cube increases, and σ_R increases proportionally. On unloading, a new equilibrium position 1″ 2″ 3″ (See Figs. 3 and 4) is reached. However, because directions 2 and 3 were only stressed elastically, the end face positions for zero lattice stresses are 1* 2 3 and the unit body must be under stress in the unloaded position 1″ 2″ 3″. $1^* 1'' = 22'' = 33''$ as the equi-axial space stress after unloading, σ_{R0}, is in each principal direction equal to the lattice stress.

$$\sigma_{R0} = \sigma_{g0} = 1^* 1'' \times E_g = e_0 E_g = e_0 \frac{m}{m+1} E \quad (19)$$

Although the body is over-strained in one direction only, its behaviour in any other principal direction is identical with that in the direction of yield. It does not matter therefore in which direction the body is over-strained, what matters is whether it has been extended or compressed.

Let the over-stretched, and then unloaded body, be again stretched in direction 1 only. The unloaded state was determined by eq. (19), after re-loading (Fig. 4), $1'' 1' = e_1 = \sigma_1/E$, $2'' 2' = 3'' 3' = -e_1/m$ and

$$\sigma_{g1} = (e_1 - e_0) E_g \; ; \; \sigma_{g2} = \sigma_{g3} = \sigma_R =$$
$$= \left(\frac{e_1}{m} + e_0\right) E_g \; ; \; \sigma_1 = \sigma_{g1} + \sigma_R \; .. \quad (20)$$

Substituting from eq. (18),

$$\sigma_{g2} = \sigma_{g3} = \frac{\sigma_1}{m+1} + \sigma_{R0} \; ; \; \sigma_{g1} = \frac{m}{m+1}\sigma_1 - \sigma_{R0} \quad (21)$$

In these calculations, yielding is assumed to occur in one

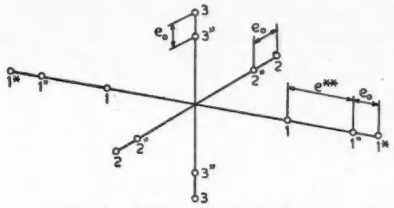

Fig. 3. **Axonometric representation of particles 1″, 2″, 3″ situated in the end faces of a unit cube, after yield in direction 1. The stress-free position of particle 1″ has been displaced from 1 to 1*.**

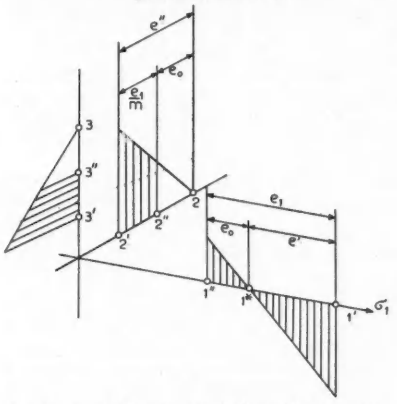

Fig. 4. **Axonometric representation of lattice stresses and particles 1′, 2′, 3′ situated in the end faces of a unit cube, after prestressing as shown in Fig. 3 and under further tension in direction 1.**

direction when the lattice stress in that direction exceeds the elastic limit. If yielding in direction 1 now progresses so far that the elastic limit is also reached in the principal directions 2 and 3, then, at this limit, $\sigma_{g1} = \sigma_{g2}$, and from eq. (20), (Fig. 4),

$$e'' = e' = e_0 + \frac{e_1}{m} = e_1 - e_0$$

$$\text{or} \quad e_0 = e_1 \frac{m-1}{2m} \quad .. \quad .. \quad (22)$$

Also at this limit the permanent unit deformation, measurable after unloading, is $-e_0$ in directions 2 and 3, and $+ 2e_0 + \Delta V$ in direction 1. Since the space stress remaining after unloading, σ_{R0}, is determined by the lattice stress, $e_0 E_g$, set up by lateral contraction (see eqs. (10) and (13))

$$e^* = 2e_0 + \Delta V = 2e_0 + 3\frac{\sigma_{R0}}{E_R} = 2e_0 + 3e_0\frac{E_g}{E_R} =$$

$$= m\,e_0 = \frac{m-1}{2} e_1 \quad .. \quad (23)$$

For a steel of 37,000 lb. per sq. in. yield strength, $E = 29 \times 10^6$ lb. per sq. in., the apparent elongation e_1 at the yield point is 0·128 per cent, and the actual elongation e^* of the material is $0{\cdot}128 \times 7/6 = 0{\cdot}15$ per cent. The 0·2 per cent proof stress lies definitely beyond the yield point.

At the moment of yielding in the lateral direction (Fig. 4), the total stresses under load are, from eq. (22)

$$\sigma_R = \sigma_{g2} = e'' E_g = \frac{m+1}{2m} e_1 E_g = \frac{e_1 E}{2} = \frac{\sigma_1}{2};$$

$$\sigma_{g1} = e' E_g = \sigma_R \quad .. \quad .. \quad (24)$$

i.e. the external uni-directional load is taken up to equal parts by the lattice stress in the load direction and by the space stress. In the strictly elastic range however :

$$\sigma_R = \sigma_{g2} = \frac{e_1}{m} E_g = \frac{\sigma_1}{m+1}$$

If in either case the same maximum lattice stress is reached, then the apparent elastic stress due to external loading is (Fig. 4)

$$\sigma_{el} = \sigma_{g1} + \sigma_R = e' E_g + \frac{e'}{m} E_g = e' E \quad .. \quad (25)$$

and the actual stress due to yielding and external loading is

$$\sigma_y = \sigma_{g1} + \sigma_R = 2\sigma_{g1} = 2e' E_g = \frac{2m}{m+1} e' E = \frac{2m}{m+1} \sigma_{el} \quad (26)$$

For steel,

$$m = \frac{10}{3}, \; \sigma_y = \frac{20}{13}\sigma_{el} = 1{\cdot}54\,\sigma_{el} \quad .. \quad .. \quad (27)$$

The internal pre-stressing has increased the yield strength by 54 per cent. On the other hand, a material whose 0·2 per cent proof stress is known to be 37,000 lb. per sq. in. has already exceeded the yield point when stressed to 0·2 per cent elongation. Its elastic limit is then 13/20 of the proof stress, i.e. 24,000 lb. per sq. in. This reduced elastic limit of a material coincides with its endurance limit and this is to be expected because the material cannot be stressed indefinitely beyond the true elastic limit if alternating stresses strain the material in extension and compression and prevent the stress-hardening effect of the space stress.

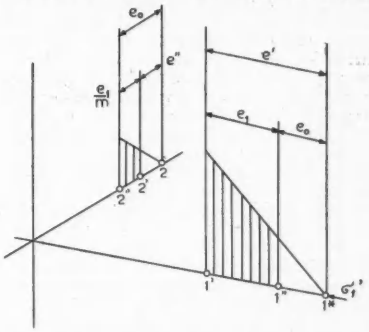

Fig. 5. **Axonometric representation of lattice stresses in directions 1 and 2 after pre-stressing by tension as shown in Fig. 3 and under compression in direction 1.**

If the over-stretched specimen is loaded in compression, then, from Fig. 5 and eq. (22),

$$\sigma_{g1} = e' E_g = \frac{m}{m+1} e' E \; ; \; \sigma_{g2} = \sigma_\rho = e'' E_g =$$

$$= \left(e_0 - \frac{e_1}{m}\right) E_g = \frac{m-2}{m+1} e' E \; ;$$

$$\sigma_1'' = \sigma_{g1} - \sigma_\rho = \frac{2}{m+1} e' E = \frac{2}{m+1} \sigma_{el} \quad (28)$$

At the elastic limit in compression, the true stress in the over-stretched material is only $2/(m+1)$ of the apparent stress due to the outer loading, and the elastic limit in compression has been reduced by $(m-1)/(m+1)$ or, for steel, by $7/13 = 54$ per cent. The elastic limit of a pre-stretched material decreases under compression by the same amount that it would increase under further tension. If the material is made to

yield under compression, a compressive space stress can be built up gradually and, under still further compression, increase the compressive elastic limit.

IV. INTERPRETATION OF EXPERIMENTS.

The calculated stress-hardening effect is known to occur with all ductile materials, stretched or compressed beyond the yield point. It disappears when the internal stresses are eliminated by slightly heating it, i.e. by recovery without recrystallization. Tests on brass[2] confirm the calculated values of yield strength increase. When yielding starts in all three principal directions at once, the stress-strain diagram shows a pronounced yield point. On load reversal, yielding first starts in one direction, the space stress is eliminated and a reversed space stress is built up. The diagram therefore shows gradual curving[3]. The shift of the elastic limit starts at very small deformations[4] and reaches its maximum value at 0·1 to 0·5 per cent deformation, i.e. at the maximum space stress value when yielding starts in all three principal directions. In repeated tensile tests above the endurance limit (the start of uni-directional yielding), a step-by-step increase in yield strength has been found up to a maximum value[4]. This increase corresponds to the building up of space stress due to uni-directional yielding.

Stress-hardening can raise the yield strength by about 35 per cent. For $m = 10/3$, $\sigma_e = 13/20\ \sigma_y$, i.e. the true elastic limit is 7/20 or 35 per cent lower than the apparent limit (say 0·2 per cent proof stress). Further work-hardening up to the ultimate strength is not due to pre-stressing but to a further increase in lattice stress. The transverse yield is at first about half the axial yield but increases at a relatively faster rate. The lattice stresses probably tend towards a maximum possible value which is first reached by the axial lattice stresses.

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SWEDEN

The Ignition of Explosive Gas Mixtures by Low-Voltage Sparks

By D. MÜLLER-HILLEBRAND. (From *Teknisk Tidskrift*, Vol. 77, No. 41, 1947, pp. 761-765, 3 illustrations.)

EXPLOSIVE gas-air mixtures can be ignited by low-energy electric sparks. Most investigations into the ignition process and the many factors affecting it have been carried out by British research workers, but the problems involved are not yet clarified, neither from the physical nor from the technical point of view. This article is a contribution to the information on the physical factors bearing upon the probability of ignition, and the precautions necessary in explosive atmospheres to preclude ignition by sparks such as may occur in the operation of the circuits of relays, signalling apparatus, and emergency lighting systems.

Of gas-air mixtures encountered those of hydrogen and air are particularly prone to ignition by an electric spark, but the ignition of hydrocarbon-air mixtures such as ethane-air, propane-air, or butane-air requires a much higher spark energy. With certain gas-air ratios the tendency towards ignition is particularly pronounced, and in such mixtures the oxygen content is so large that it combines with the gas with a maximum of heat generation. The ignition temperature has not a fixed value, however, but chemical reactions at the surface of the heated parts present (i.e. catalytic effects), and intermediate products of reaction, etc., play an important role. If heated bodies are introduced into an explosive mixture, the ignition temperature will be higher as the surface areas of these bodies are smaller.

The velocity distribution of the gas molecules at a given temperature is defined by Maxwell's law of velocity distribution ; molecules with higher velocities will be rare. The most important requirement for the occurrence of an explosion is that some molecules must be able to act as carriers of the activating energy owing to, for example, their high velocity, so that the reaction can be started. The chemical transformations will commence with a few activated molecules, and their kinetic energy, when colliding, must be at least equal to the activation energy. Furthermore, collision must take place in a certain manner but according to Maxwell's velocity distribution only a very small number of molecules will satisfy these conditions and, consequently, only a minute portion of the thermal energy supplied s expended upon initiating an explosion. In other words, if an explosion is to be caused by heat, then the total energy which must be expended in generating this heat is many times more than the amount of energy necessary to cause the ignition. If it is possible to obtain activated molecules by some other process with higher efficiency, then the energy required for ignition will be smaller. This applies to electric sparks, or, more precisely, to a part of the electric spark discharge process.

The electric spark cannot be simply considered as a " hot cylinder " in which the dissociation occurring at the prevailing high temperature leads to the production of free atoms and radicals. Actually the molecules are activated not only by the heat release but also by electrical effects. Considered as a source of energy the spark is more powerful than a hot cylinder and the energy involved in the spark effect can be qualitatively more effective than the energy which causes ignition by a thermal effect alone. Nevertheless one must take into consideration that in the case of very short sparks the greater part of energy is supplied as heat by conduction to the electrodes and that only a small portion is supplied to the gases by diffusion and radiation in the form of qualitatively highly effective energy. Also, in the case of very short sparks the metal vapour present exerts a greater retarding influence upon the occurrence of an explosion than is the case with longer sparks.

In a closed contact the electric current flows largely through one or several exceedingly small areas of metal, the lines of current flow concentrating at these points. As the contacts begin to separate with incipient interruption the contact areas are diminished in size, and the current density and heating effect is increased. The heated area is so small, however, that temperature equilibrium is established almost instantaneously, the actual time, for example, being less than one microsecond (10^{-6} sec.) in the case of a 10 amp. contact. The electrodes will melt and become joined over a small area of liquid metal, and when the electrodes are separated, the surface forces exerted upon the droplet of liquid metal causes it to be elongated as a bridge between them. With silver electrodes the hottest point of this bridge will be at a temperature somewhat below

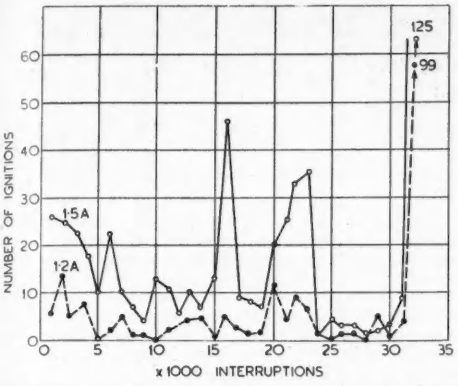

Fig. 1. **Number of interruptions *versus* number of ignitions for 1·2 and 1·5 amp., respectively. Inductance 10^{-3} Henry, voltage 6 V.**

the boiling point of silver which is 1950 deg. C. It is the opinion of the author that despite this high temperature it is improbable that ignition of a hydrogen-air mixture will be caused (although hydrogen and oxygen react) because the temperature gradient, and the heat dispersal, is so large that an incipient reaction cannot develop to the point where ignition occurs.

When the bridge between the electrodes is further elongated it becomes restricted at the hottest point, the surface forces being effectively assisted by the magnetic force. The magnetic field produced by the current flowing through the bridge, owing to the high current density of 10^6 amp. per sq. mm. creates a high internal pressure, which at the instant of interruption may reach several hundred atmospheres. This leads to an increase in the vaporizing temperature of the silver contacts to more than 3,000 deg. K., or even to 4,000 deg. K. if the current does not diminish. This high temperature creates the conditions for the existence of an arc between the contacts. The energy required for the evaporation of silver will be available for a very short period which is in the order of 10^{-7} seconds. In a bridge between silver contacts carrying 10 amp., the energy flux will amount to as much as 260 million kW per cm³. But it is probable that the short interruption period of one microsecond, attended by evaporation of the metal, is not sufficient to cause ignition, at least if the current is less than 30 amp. Only the gas discharge following the interruption period will lead to ignition. In the case of an initial current of 10 amp. the bridge is broken when the distance between the contacts has increased to 2-3 microns. Whether the amperage will have decreased at this moment will chiefly depend upon the inductance of the circuit. If the current passing through the silver contacts is larger than 0·4 to 0·9 amp., there will be an arc the voltage of which may be between 12 and 24 volts. But if the current is smaller, the discharge will be quickly transformed into a glow discharge the voltage of which may reach 300.

For the experimental investigation of the ignition current a contactor was installed in a steel vessel of 0·5 litres volume. An explosive mixture of 22 per cent hydrogen-air mixture was chosen which is particularly easy to ignite with the electric spark. The mixture ratio was kept constant during the tests and was checked by using an interferometer. The contacts consisted of 2 mm. dia. pins of fine silver with rounded ends. Prior to the test the contact surfaces were polished with emery cloth and cleaned with alcohol.

A 6 volt storage battery was used and the contact was interrupted four times per second, polarity being reversed after every ten interruptions to avoid the building up of metal deposits on the electrode tips.

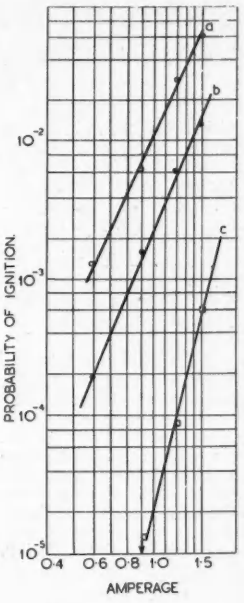

Fig. 2. **Probability of ignition *versus* amperage. Inductance 10^{-3} Henry, Voltage 6 V.**
(a) 4,000 interruptions;
(b) 32,000 interruptions;
(c) 68,000 interruptions.

Two series of tests were run with an inductance of 1 mH in the circuit. The first of these comprised 32,000 interruptions and was remarkable for the extraordinarily high differences in the igniting power of the sparks (see Fig. 1.) With 1·5 amp. as well as with 1·2 amp. the number of ignitions was found to decrease over the first 10,000 interruptions but then to increase suddenly so that, towards the end of the test, the number of ignitions had increased to a hundred times as many.

From Fig. 2 it will be seen that the probability of ignition increases with the 4th to 8th power of the amperage. This rule applies to probabilities of less than 5×10^{-3} generally based on inductances between 10^{-6} H and 1·0 H and for current intensities between 30 and 0·1 amp. Fig. 2 shows the wide range over which the ignition probability may vary, this chart being based upon tests conducted with two electrode sets subjected to 32,000 and 68,000 interruptions respectively.

Although the tests referred to above are incomplete, it can be concluded that the gas discharge between the contacts caused by interruption of the circuit is responsible for the occurrence of ignition. Circuits with purely ohmic resistance are frequently referred to in literature but rarely exist in reality, and an inductance as low as 10^{-6} H can cause a short-time gas discharge. Test data of this nature should therefore include information with regard to circuit inductance.

Another conclusion drawn by the author is that, when breaking contact in circuits the e.m.f. of which is considerably below the gas discharge potential, only a very small portion of the spark energy is employed for ignition.

Taking an ignition probability of 10^{-4} for hydrogen-air mixtures as a basis, an estimate can be made of the amperage of interruption which can be considered safe. These amperages are given in the following table :—

Maximum permissible current, amp.	Maximum permissible inductance, Henry	Voltage
6·5	10^{-5}	up to 6 V.
2·0	10^{-4}	
0·55	10^{-3}	
0·3	7×10^{-3}	up to 48 V.
0·2	2×10^{-2}	
0·1	9×10^{-2}	

U.S.A.

Electroless Plating on Metals by Chemical Reduction

(From *Technical News Bulletin, National Bureau of Standards*, Vol. 31, No. 10, October, 1947, pp. 111-112, 1 illustration.)

A NEW process for plating nickel and cobalt on metal surfaces without the use of electric current, developed by Abner Brenner and Grace E. Riddell of the Bureau's electroplating laboratory, is brought about by chemical reduction of a nickel or cobalt salt with hypophosphite in hot solution. The reaction is catalytic, and under the prescribed conditions of concentration and pH, no plating occurs unless certain metals, such as steel or nickel, are introduced in the bath. The reduction then occurs only on the surface of the immersed metal with the production of an adherent coating of 93 to 97 per cent purity.

Photomicrographs of deposits obtained in this way show both a laminar and a columnar structure, similar to bright nickel electrodeposits. The electroless deposits are of good quality—sound though brittle, and usually bright. As they can be made as hard as tool steel, the method may prove useful where hard, wear-resistant surfaces are required, as in bearings. The process is particularly applicable to the plating of recesses, irregular-shaped objects, and enclosed areas such as tubes, where a centred internal electrode with special current leads would be needed in electroplating.

The equipment is simple and more easily assembled than that required for electroplating. No generators, rheostats, special racks, or contacts are necessary. Small parts that cannot be barrel-plated economically are readily plated by the electroless process if suspended by a string or in a bag affording ample exposure of the metal surface to the solution. There is no need of constant motion, as in barrel plating, as current distribution is not involved.

Although electroless deposits of cobalt and cobalt-nickel alloys have been obtained only from ammoniacal solutions, nickel can be deposited from either acid or alkaline solutions. The reactions, requiring a temperature of about 90 deg. C., are given in the following equations (in which cobalt may also be written for nickel) :

$$NiCl_2 + NaH_2PO + H_2O \rightarrow Ni + 2HCl + NaH_2PO_3$$

or

$$NaH_2PO_2 + H_2O \rightarrow NaH_2PO_3 + H_2.$$

The first reaction is the important one, resulting in the deposition of nickel ; the second reaction tends to lower the efficiency of the process through oxidation of the hypophosphite.

A unique feature of the electroless process is the catalytic initiation of the reaction by the following metals : Iron or steel, nickel, gold, cobalt, palladium, and aluminium. Unless one of these metals is introduced into the solution, no reaction takes place. Once started, the reaction continues at the metallic surface and only rarely occurs in other parts of the bath. For this reason, the containing vessel should be of glass. plastic, or other noncatalytic material.

Objects to be plated are cleaned and given an acid dip before being suspended in the hot solution. The rate of deposition is about the same as in barrel electroplating, depending upon the type of solution used. During the process, the pH must be kept within a certain range, and if the operation is of long duration, the nickel salt and hypophosphite are replenished at regular intervals.

The composition of the plating bath may vary within rather wide limits. Suggested solutions are given in the table. In addition to the metal ions and hypophosphite, the alkaline solutions contain hydroxycarboxylic salts and ammonium salts to prevent precipitation of the metal salts, and certain alkalies to regulate the pH.

Electroless plating on noncatalytic metal surfaces may be accomplished in two ways. If a film of palladium or rhodium of nearly monatomic thickness is first applied by chemical replacement on a noncatalytic metal, deposition of nickel or cobalt will occur on the activated surface. Electroless plating of copper may be carried out in this way. A second method of initiating the reduction is to bring a less noble metal, such as iron or aluminium, in contact with the noncatalytic metal while it is immersed in the hot electroless solution. Once the process has been started, it continues because of the catalytic action of the initial deposit.

Suggested compositions for the various electroless solutions.

	Nickel		Cobalt	Cobalt-Nickel
	Alkaline	Acid		
	g/litre	g/litre	g/litre	g/litre
Nickel chloride, $NiCl_2.6H_2O$	30	30	—	30
Cobalt chloride, $CoCl_2.6H_2O$ ..	—	—	30	30
Sodium hypophosphite, $NaH_2PO_2.H_2O$ ..	10	10	20	20
Sodium citrate, $Na_3C_6H_5O_7.5\frac{1}{2}H_2O$ ..	100	—	35	—
Rochelle salt, $NaKC_4H_4O_6.4H_2O$ ..	—	—	—	200
Sodium hydroxyacetate, $NaC_2H_3O_3$	—	50	—	—
Ammonium chloride, NH_4Cl	50	—	50	50
Alkali for neutralizing ..	NH_4OH	NaOH	NH_4OH	NH_4OH
pH	8 to 10	4 to 6	9 to 10	8 to 10
Rate of deposition, in./hour	0·0003	0·0006	0·0006	0·0006
Appearance of deposit ..	Bright	Semi-bright	Dull	Semi-bright

As formed, the electroless nickel deposits are brittle but become ductile when heated. These deposits are harder than the ordinary electrodeposited nickel, and upon heating their hardness is still further increased. This is in contrast to the behaviour of " hard " electrodeposited nickel, which has high initial hardness, but softens upon heating. The hardening may be explained as " precipitation hardening," probably of phosphides.

The adhesion of the nickel deposit to mild steel is such that it cannot be flaked off by bending, but on high-carbon steel this property is less satisfactory. In salt-spray tests on steel coated with 0·0002, 0·0005, and 0·001 inch of electroless nickel, in comparison with similar panels coated with electrodeposited nickel, the protective value of the two types of coatings was virtually the same. However, the electroless cobalt does not compare so favourably with electrodeposited cobalt.

The yield or efficiency of the reduction based on the decomposition of hypophosphite is 37 and 66 per cent for nickel and cobalt, respectively. About 2 grams of nickel or nearly 4 grams of cobalt are reduced by 10 grams of sodium hypophosphite. Because of the moderate yield and the present high cost of sodium hypophosphite, the process is expensive. Extensive commercial use of electroless plating is thus dependent upon a reduction in the price of this chemical.

Wear and Defects of Engine Bearings

By F. PICARD. (From *Le Génie Civil*, Vol. 124, No. 12, June 15, 1947, pp. 233-235, 4 illustrations).

IN its first report on the crankshaft and connecting rod bearings of I.C. engines, the Sub-committee on Bearings, of the French National Scientific Research Centre, has determined a new coefficient of wear and more closely defined certain specified defects of bearings.

I. COEFFICIENT OF WEAR.

The coefficient of wear determines the qualities of a bearing working under a certain load and shaft speed, gives a comparison of the behaviours of different anti-friction alloys, establishes the most favourable running conditions for each bearing material, and forms a useful basis for the calculation of bearing dimensions. The mechanical factors governing bearing behaviour are usually considered together and characterized by the energy dissipated in frictional heat in the bearing.

$$P_f = \mu \,.\, F \,.\, V = \mu \,.\, p \,.\, A \,.\, V = \text{prop. } p \,.\, V,$$

where μ = the friction factor, p = the pressure per unit bearing area, V = the rubbing speed. However, numerous tests have shown that the rubbing speed V has a greater influence on bearing resistance and wear than the pressure p, especially with cupro-lead bearing alloys. The French Centre for the Study of Bearing Problems has therefore suggested that the wear coefficient $p \,.\, V$ may now be modified by assuming that bearings normally run under hydrodynamic friction conditions and that consequently :—

$$P_f = u\, p_m\, V = 3{\cdot}8 \sqrt{\frac{\eta\, V}{r\, p_m}} \times p_m \times V =$$

$$= \frac{3{\cdot}8}{\sqrt{r}} \sqrt{\eta}\, \sqrt{p_m V^3} = \text{prop. } \sqrt{p_m V^3}$$

where p_m = mean bearing pressure, η = viscosity of lubricant, r = bearing radius. The new coefficient of wear $\sqrt{p_m V^3}$, for η = const., has been found to agree better with experimental results and has been generally adopted by the Sub-committee. It does not, however, make allowance for the particular influence of maximum pressure and of load fluctuations which certainly affect bearing resistance and wear. To study this influence, the Sub-committee has distinguished between the following three cases and more closely investigated the conditions under (3) :—

(1) Bearings under constant load ;
(2) bearings subject to medium load fluctuations, such as crankshaft bearings of I.C. engines ;
(3) bearings subjected to severe load fluctuations such as connecting rod bearings of I.C. engines.

For every crank and connecting rod position, a polar diagram can be drawn of the combined force F exerted on the crank, by the gas pressure, the inertia effect of oscillating and the centrifugal effect of rotating masses. This diagram permits calculation of the mean load F_m and of load fluctuations, and determines the best position for oil holes. For a simple comparison of the maximum running conditions of different engines, cylinder pressure can be neglected with respect to inertia and centrifugal forces, and it is then sufficient to use the following approximate formula (see Fig. 1) :—

$$F_m = 0{\cdot}665\, R\omega^2 \sqrt{m^2\left(1+\frac{1}{\lambda^2}\right) + 2m'^2 + mm'\left(2+\frac{1}{\lambda^2}\right)} =$$

$$= 0{\cdot}665\, MR\omega^2$$

where F_m = the mean bearing load, R = the crank radius, ω = the mean angular velocity, m = the oscillating mass

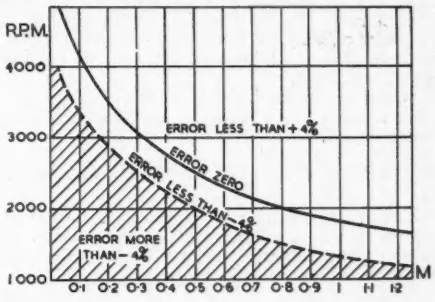

Fig. 1. **Approximation obtained, for different values of M and ω, by application of formula : $F_m + 0{\cdot}665\, MR\, \omega^2$.**

(piston and small end), m' = the rotating mass (big end of conn. rod), λ = the ratio of conn. rod length to crank radius, M = the equivalent mass of moving parts (excepting crankshaft). The instantaneous speed is

$$v = V_m\left(1 + \frac{\cos\alpha}{\sqrt{\lambda^2 - \sin^2\alpha}}\right)$$

where V_m = the mean velocity and α = the angular crank position.

V^3 is then the mean value, taken over one revolution, of the cubes of the values v, but this calculation is rather tedious and it is generally sufficient to take the cube of the mean velocity V_m and to correct results by multiplying by a coefficient ϵ which is only dependent on λ (see Fig. 2). With

$$p_m = \frac{F_m}{LD} = \frac{0{\cdot}665 MR\omega^2}{LD} \text{ and } V_m = \frac{\omega D}{2}$$

where D = the bearing diameter, L = the true bearing length. A final coefficient of wear for these conditions is

$$\epsilon \sqrt{p_m V_m^3} = \epsilon D \sqrt{\frac{0{\cdot}665 MR}{8L}}\, \omega^5$$

which increases with the 2·5th power of the angular velocity

After extensive studies of the different running conditions of engines, the Sub-committee decided that calculations of the coefficient of wear should be based on the following conditions :—

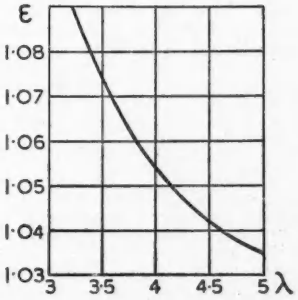

Fig. 2. **Correction factor ϵ as a function of connecting rod ratio λ.**

(1) For aircraft ; the conditions at take-off and at cruising speed.

(2) For automobiles; the maximum and average running conditions. Average running conditions are determined, for a car with three forward gears, by the average car speed in direct gear. For cars with less than 90 cub. in. cylinder capacity, they are about 70 per cent, and for more powerful cars, about 60 per cent of maximum running conditions.

(3) For marine and industrial engines for rail and road traction ; the running conditions at normal governor speed.

II. DEFINITION OF CERTAIN BEARING DEFECTS.

Seizing. A defect in the form of numerous deep scratches in the direction of shaft displacement. Not observed with anti-friction bearing materials of low melting point (tin or lead base).

Superficial fusion. Metal surfaces slightly fused at certain parts of the bearing. Observed on materials of low melting point only.

Running out. Bearing material completely molten and run out of bearing bush. Observed on materials of low melting point only.

Cracks. Cracks parallel to bearing length or, in an advanced stage, criss-crossed, " mosaic " cracks. Observed on anti-friction materials of low melting point and on cupro-lead alloys.

Pitting and decay. Deep grooves concentrated in certain bearing regions. " Pitting " if grooves are isolated, " decay " if grooves are numerous and concentrated. Observed on cupro-lead alloys only.

Peeling. Anti-friction material peeling in flakes from its supporting material, generally at zones where decay or mosaic cracks develop. The backing metal is not entirely exposed but covered by a thin adherent layer only. Observed on cupro-lead alloys.

Scratches. Isolated scratches only. This defect is very different from " seizing " and occurs with anti-friction materials of low melting point and with cupro-lead alloys.

The committee invites comments, suggestions, and communications about experimental results.

HOLLAND

The Practical Construction of Vibration-Free Mountings with Auxiliary Mass

By J. A. HARINGX. (From *Philips Technical Review*, Vol. 9, No. 3, 1947, pp. 85-90, 7 illustrations.)

IN order to reduce the amplitude of the forced vibrations of a sensitive instrument situated in the vicinity of vibrating structures, it can be placed on an elastic mounting. This mounting, however, must be designed so that its damping will ensure a sufficiently rapid decay of the free vibrations of the system which may be caused by slight initial displacements or impulses. In a previous article[1] the author showed that there are certain advantages obtainable by introducing the damping between the apparatus and the auxiliary mass connected to it by means of springs, and he investigated the features of the system for the case of one-dimensional motion.

In practice, the conditions are more complicated, because the foundation is generally capable of vibrations in different directions and about various axes. The mounting should therefore be resilient in several directions, but these directions, and the corresponding centres of the elastic forces, cannot be arbitrarily chosen, because the vibratory motion of the system might then become highly complicated and impossible to determine, and a more detailed consideration of the principles involved is necessary.

A rigid body suspended in space by springs has six degrees of freedom, for translatory motion along, and rotary motion about, three orthogonal axes. These degrees of freedom are generally coupled together so that, for example, motion in the direction of an applied force is the cause of translatory and rotary motion in other directions also. The coupling between the degrees of freedom can be eliminated, however, by designing the sprung mounting so that it will have a *centre of elasticity* defined as the intersection of the three principal axes of elasticity which are orthogonal to each other. These axes are chosen to coincide with the principal axes of the suspended body, so that its centre of gravity will also be the centre of elasticity.

A *principal axis of elasticity* is characterized by the fact that a force acting in its direction causes translatory motion in this direction only, while a couple about this axis causes rotation only about this same axis. Under these conditions, the system has, in effect, six non-coupled (i.e. independent) degrees of freedom. A system consisting of an arbitrary combination of elastic elements does not usually possess a centre of elasticity. A mounting can be designed to meet this requirement, however, if it is symmetrical with respect to two planes perpendicular to each other (as in Figs. 1a, b, c) and if its elastic elements are equally stiff in directions perpendicular to these planes. The z-axis, which is the intersecting line of the two planes, must then, for reasons of symmetry, be one of the principal axes of elasticity, while the other two principal axes must be parallel to the x– and y–axes. The position of the z–axis can be calculated from the spring stiffness.

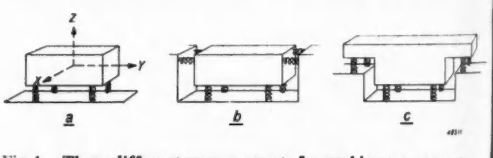

Fig. 1. **Three different arrangements for making an apparatus vibration-free where multi-dimensional disturbing vibrations exist. The mountings are all symmetrical with respect to two perpendicular vertical planes.**

Provided conditions of symmetry in two directions are complied with, the vibrating system will have six degrees of freedom which (disregarding the disturbing influence of gravity) are independent of each other, and its vibrations can then be treated as six separate one-dimensional problems.

The vibration-free mounting given in Fig. 4 of the previous article on this subject[1] also complies with these requirements. When damping elements are used, they must be arranged so that in the case of translatory displacements parallel to, or rotations about, the three principal axes, the damping will result in forces or couples whose vectors are in those same directions. In practice these conditions can be satisfied with sufficient accuracy, and the problem is then reduced to choosing the parameters for each degree of freedom separately so that the system is as free as possible from vibrations.

DESIGN OF SPRINGS.

Mechanical disturbances may be caused by the regular throbbing of a machine or they may be due to all kinds of irregular shocks such as footsteps and the slamming of doors. Consequently the frequency spectrum of the disturbances will also contain very low frequencies. The vibrations are first transmitted by the floors, walls and tables of buildings, each of which acts as filter transmitting vibrations mainly in the neighbourhood of its resonant frequency. As a result of an analysis of such vibrations with the aid of a vibrometer, it was found that the lowest natural frequencies occurring in practice are usually between 15 and 20 cycles per second. To make the mounting insensitive to these vibrations of the foundation, it must be designed to have a very low resonant frequency, of 2-4 cycles/sec.

These resonant frequencies are so low that they preclude the use of rubber cylinders for the resilient elements of the mounting, because they would be too rigid. The resonant frequency ω_0 of a system is given by $\omega_0 = \sqrt{g/f}$, where g is the acceleration due to gravity and f the deflection (due to the compression of the resilient element caused by the weight of the mass), so that to obtain a resonant frequency $\omega_0/2\pi$ of, say, 3 c/sec. f must be 2·8 cm. Rubber cylinders would have to be at least 10 cm long to give the required deflection ; and to prevent buckling, their diameter D would have to be of the same order of magnitude, so that the force necessary for the compression would be of the order of 1,000 kg, according to the formula $P = \pi D^2 Ef/l$, assuming the modulus of elasticity $E = 10$ kg/cm² for the weakest kind of rubber available. A mounting with four such cylinders would weigh over 4 tons. Sponge rubber cylinders would enable smaller compressive forces to be used, but this is not a very durable material and it is not recommended for this purpose.

Helical steel springs are much more suitable, and can easily be designed to give a deflection of about 3 cm under a load of about 10 kg. The natural frequencies of these springs are usually so high that they do not cause any trouble in mountings of the type being considered. Furthermore, helical steel springs possess not only axial, but also lateral, rigidity, provided they are not hinged at both ends, and this lateral rigidity can also be profitably used in the vibrating system to make possible a much simpler design (see Figs. 1a and 1c).

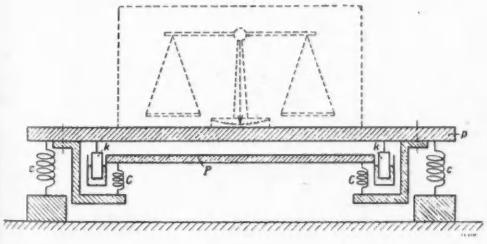

Fig 2.

The practical arrangement is shown in Fig. 2. The flat plate p, carrying the sensitive instrument, is supported by four helical springs. The auxiliary mass, P, is a second plate, supported by the first in a similar manner. In both cases the springs act as resilient elements both in the vertical and in the horizontal directions.

The lateral rigidity of the springs can be calculated by means of the formula

$$c = 4\,m\,EI/[(m+1)\,\pi\,n\,D^3]$$

where E = modulus of elasticity, I = moment of inertia of the circular cross-section of the wire, n = = number of turns, D = coil diameter, and m = = Poisson's ratio = 10/3 for steel. The theory of the lateral deflection and the danger of buckling of helical springs has also been investigated by the author[2] and the following is a brief review of the results.

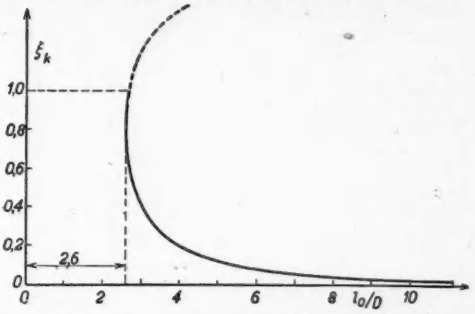

Fig. 3. Critical compression ξ_k as a function of the slenderness ratio l_0/D of the spring. This relation is valid for a spring with both ends hinged or remaining parallel. When both ends of the spring are fixed, l_0 is to be taken as half the length of the spring. Buckling cannot occur as long as $l_0/D < 2{\cdot}5$.

A slender helical spring will buckle as soon as the compressive load, and hence also the relative compression ξ, has reached a certain critical value ξ_k which depends, at least in a first approximation, only on the ratio l_0/D, which is called the *slenderness ratio* of the spring (l_0 = length of the unloaded spring, D = spring diameter). A more precise calculation shows that the pitch and the wire diameter also have a slight effect. The relation found for the case where both ends of the spring are hinged or remain parallel after deflection (as in Fig. 2) is shown in Fig. 3. It will be seen that the spring cannot buckle when $l_0/D \ll 2{\cdot}6$, and this condition must therefore be taken into account when designing springs.

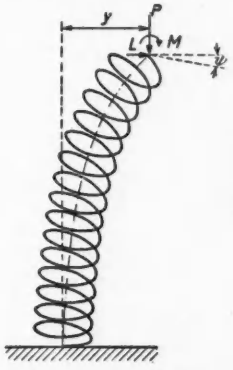

Fig. 4. Lateral deflection y and rotation ψ of free end of a helical spring subjected to a transverse force L, a couple M, and compressed axially by a weight P. The lower end of the spring is clamped.

Further, the lateral deflection y of the free end of an axially compressed spring under the simultaneous influence of a lateral force L and a couple M (see Fig. 4) is given by the equations

$$\left.\begin{aligned} L &= c_1 y - c_2 \psi \\ M &= -c_2 + c_3 \psi \end{aligned}\right\} \quad \ldots \quad \ldots \quad (1)$$

where ψ is the rotation of the upper end, and c_1, $_2$, $_3$ are the stiffness coefficients. These coefficients are related to the axial stiffness c by the following equations :—

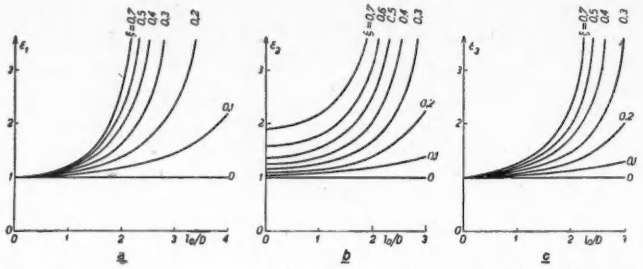

Fig. 5. **The correction factors ϵ_1, ϵ_2, ϵ_3 occurring in formulae (2) as functions of the ratio of slenderness l_0/D with the compression ξ as parameter.**

$$\frac{c}{c_1} = \frac{m}{2\,(m+1)}\left[1 + \frac{2m+1}{3m}\left(\frac{l}{D}\right)^2\right]\epsilon_1 \qquad (2a)$$

$$c_2/c_1 = l\,\epsilon_2/2 \quad .. \quad .. \quad .. \qquad (2b)$$

$$\frac{c_3}{c_1} = \frac{m\,D^2}{4\,(2m+1)}\left[1 + \frac{4\,(2m+1)}{3m}\left(\frac{l}{D}\right)^2\right]\epsilon_3 \qquad (2c)$$

In these formulae l is the length of the spring when loaded, and $\epsilon_{1,\,2,\,3}$ are correction factors depending on ξ and l_0/D, and are represented graphically in Figs. 5a, b, and c.

By applying these results to the mounting shown in Fig. 2, the height h of the centre of elasticity above the base plane of the springs can be calculated.

A horizontal force L through this centre will therefore cause only a lateral displacement y of the plate, which then remains horizontal, so that $\psi = 0$. Hence the couple M due to L is

$$M = -c_2 L/c_1,$$

but this couple is also given by $M = -(l - h)L$, therefore

$$h = l\,(1 - \epsilon_2/2).$$

The centre of gravity of the mounting must therefore be at this height. Furthermore, $L = c_1\,y$ so that eq. (2a) gives directly the stiffness c_1 of the spring required to make the mounting vibration-free as regards transverse vibrations.

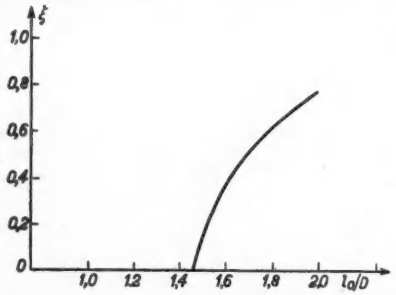

Fig. 6. **Relation between ξ and l_0/D, assuming the spring has the same axial and lateral rigidity.**

It is often desirable for the mounting to behave in the same way in the vertical and in the horizontal direction. When vertical springs alone are used, c_1 must be equal to c. This gives a relation between ξ and l_0/D which can be determined from eq. (2a) and Fig. 5a, and which is represented graphically in Fig. 6. With this graph, and with a knowledge of the load and resonant frequencies required, the conditions which have to be satisfied by the spring, and hence its length, diameter, and number of turns, etc., are fully determined.

The rigidity of the mounting under the influence of a couple about one of the principal axes can also be expressed by using the stiffness coefficients c, $c_{1,\,2,\,3}$. In this connexion, however, the contributions of the coefficients $c_{1,\,2,\,3}$ can often be neglected. For rotary motion, the same considerations as for translatory vibrations are valid, after substituting the moments of inertia for the masses.

PRACTICAL APPLICATION OF DAMPING.

A suitable damping is best achieved by using four cups k filled with oil of a given viscosity which are fixed to the auxiliary mass (plate P in Fig. 2), and in which four other cups attached to the main mass (plate p) can move freely in all directions, with a few millimetres clearance.

If it is required that the damping coefficient k should be the same in both the horizontal and vertical direction, the cups must have a square cross-section, and according to calculation they must be filled to a height $H = 1{\cdot}45B$, where B is the length of a side of the base of the inner cup. The damping coefficient is then approximately

$$k = 18\eta\,\frac{B^4}{d^3}\left[1 + 2{\cdot}5\,\frac{d}{B}\right],$$

where d is the clearance between the walls of the two cups, and η the viscosity of the liquid.

Optimum damping can be determined experimentally by using oils with different viscosities, but there is actually no great sensitivity in this adjustment, so that an approximate adjustment for damping is sufficient, and the variation of viscosity with temperature also has only a negligible effect on the behaviour of the system.

Instead of liquid damping, frictional damping can also be used to dissipate the kinetic energy transmitted to the mounting by outside disturbances. For example a set of flexible strips can be attached to one of the plates in Fig. 2 with a slight pressure against the other plate so that friction occurs owing to the relative displacement. If the amplitudes are sufficiently large, this simple arrangement will ensure a fairly rapid dissipation of the kinetic energy. However, when the amplitudes are less than a certain critical value, the frictional forces operate in a different manner, and cause one mass to follow the movements of the other. Thus, as soon as frictional damping has reduced the amplitudes to this low value, it ceases to be effective, and the instrument will continue to vibrate at this amplitude for some time. If, as is usual, the resonant frequency of the system is low, this may not constitute an objection for some instruments (such as dial gauges, microscopes and not too sensitive balances) but for galvanometers and other highly sensitive instruments the use of liquid damping will be essential.

A vibration-free mounting designed according to this theory and weighing 110 kg. can be used as a mounting for instruments up to 35 kg, and its resonant frequency is about 3 cycles/sec. This arrangement fully answered expectations both as regards the decay of the free vibrations and as regards the amplitudes of the forced vibrations.

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(1) J. A. Haringx. Vibration-free mountings with auxiliary mass. *Philips Techn. Rev.*, Vol. 9, No. 1, 1947, pp. 16-23; *Engrs. Digest*, Dec., 1947, pp. 396-399.

(2) J. A. Haringx. On the buckling and lateral rigidity of helical compression springs. *Proc. Kon. Ac. v. Wet. Amsterdam*, Vol. 45, 1942, pp. 533-539 & 650-654.

Stampings from Molten Non-ferrous Metals

By N. A. Sokolov. (From *Vestnik Inzhenerov i Tehnikov*, No. 9-10, 1946, pp. 301-306, 12 illustrations.)

Outline of Process

The stamping of molten metal differs from ordinary die casting or pressure castings in that the pressure is developed in and by the die itself, and not in an external pressure chamber. This eliminates the principal defect of die casting; i.e. the porosity which results from the flow of metal in a high velocity stream from the pressure chamber into the die.

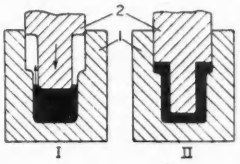

Fig. 1.

1—Matrix. 2—Plunger.
I—Plunger just touching the metal.
II—Plunger in final position.

The process is illustrated diagrammatically in Fig. 1; the requisite quantity of molten metal is first introduced into the matrix and *then* the plunger or male die descends and forces the metal into the required final shape, ensuring sound metal, accurate shape and clean surfaces. The stamping machine requires two operating cylinders at right angles: one to open and shut the matrix, the other to actuate the plunger.

Suitable Applications

The most suitable applications of the process is for bearing bushes and similar components with walls $\frac{1}{8}$ to $\frac{3}{8}$ in. thick, outside diameters up to 5 in. and length-to-diameter ratios up to 1·25. The shape of the internal surface must be such as to permit of withdrawing the plunger. Fig. 2 shows typical applications.

Fig. 2.

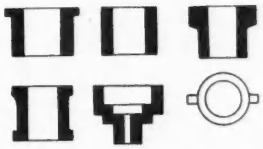

Fig. 3 shows the economy of metal effected as compared with machining from a forged blank, from a centrifugal casting and from an ordinary hot stamping; moreover, 98 per cent of the molten metal is left in the finished stamping, and the loss by splashing and fin formation is only 2 per cent.

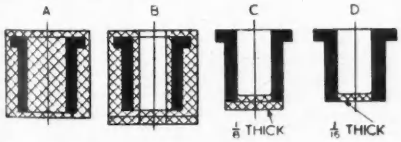

Fig 3. **Bush production from:—**
(A) Solid metal blank. (B) Centrifugal casting.
(C) Hot stamping. (D) Molten-metal stamping.

Plunger Pressure

The pressure required in the metal depends on the size of the component, as shown in Fig. 4, which, however, gives the minimum values. To cover differences in temperature, etc., it is best to choose values

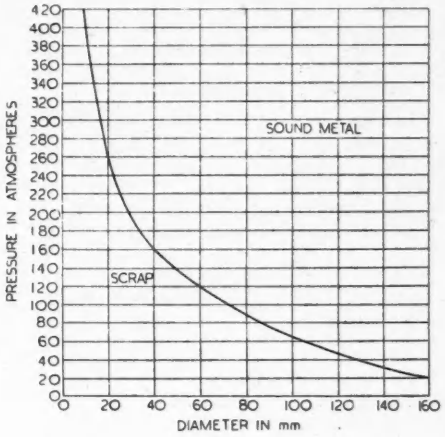

Fig. 4.

1·5 to 2 times greater than those read from the curve. If the pressure chosen is P (lb./sq. in.) and the area of the plunger is A (sq. in.) the plunger cylinder must obviously be designed to exert a force $F = P \times A$. Excess pressure does no harm but does not lead to any improvement except in so far as it can act as a corrective for incorrect temperature or slow timing of the process.

Temperature of Metal

One essential of the process is that the metal should not begin to set before pressure is applied, and therefore the initial temperature of metal and matrix and the speed of application of the pressure must be carefully studied. Excessive heating of the metal can lead to defects because of the increased amount of shrinkage and the longer cooling time.

The best initial temperature of the metal varies with its composition and corresponds roughly to the temperature suitable for castings of 1 in. thickness made in metal moulds. It is a little lower for thick stampings than for thin ones.

Temperature of the Mould and Timing of the Plunger Action

If the matrix is cold or the pressure is applied too late the stamping will exhibit the type of defect usual with metal mould castings, i.e. unsatisfactory definition, external scabs due to air pockets, and deep inclusions of oxidized skin. These defects are usually to be found in the lower part of the stamping whilst the upper part will be comparatively faultless because it is formed by the displacement of the hottest metal. Alternatively low temperature may lead to excessive thickness of the bottom of the stamping and result in the upper portion being starved of metal.

Experience shows that:

A. The matrix must be at not less than 150 deg. C. before the metal is introduced and not in excess of 400 deg. C. at the end of the process. (These temperatures can be measured by means of a thermo-couple at the surface).

B. The interval between the pouring and forming of the metal should be 0·3 to 0·4 seconds. At about 0·6 seconds small surface cracks begin to appear on the finished product.

Timing of Plunger Withdrawal, Tapering of Plunger

The improvement of metal structure is obtained essentially because it sets under pressure; therefore, pressure must be maintained until the end of the second phase of shrinkage, i.e. until crystallization is completed. Maintaining this pressure during the third phase could only be advantageous if much greater pressures were used, namely, 13 to 20 tons per sq. in.

Experiments conducted on bushes 0·2 in. thick with a plunger taper of 3 deg. have shown that if the plunger is withdrawn after one second the metal exhibited all the defects common in open mould castings. Three to four seconds invariably proved to be sufficient to make sure of homogeneous metal. Five to seven seconds result in difficulties in withdrawing the plunger. In general, the timing should be the minimum consistent with good results and in the case of extra thick and extra long bushes the taper of the plunger may have to be increased to facilitate withdrawal, in accordance with the following table.

Height to diameter ratio	Taper on each side (deg.)	
	¼ to ½ in. thickness	½ to 0·4 in. thickness
up to 0·5	2	3
up to 0·75	3	4
up to 1·0	4	5
up to 1·0	5	5

The Stamping Machine

A standard Russian machine, type OVP -2, is shown in Fig. 5 and diagrammatically in Fig. 6. It is hydraulically operated and the control box has two pairs of

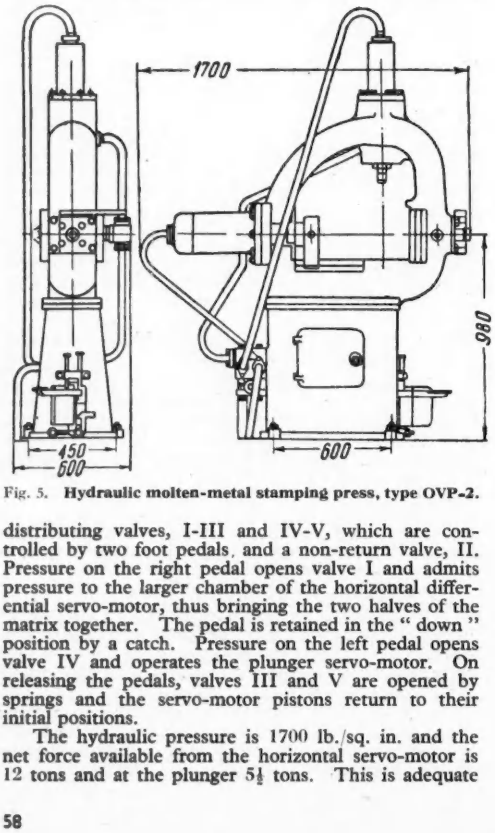

Fig. 5. **Hydraulic molten-metal stamping press, type OVP-2.**

distributing valves, I-III and IV-V, which are controlled by two foot pedals, and a non-return valve, II. Pressure on the right pedal opens valve I and admits pressure to the larger chamber of the horizontal differential servo-motor, thus bringing the two halves of the matrix together. The pedal is retained in the "down" position by a catch. Pressure on the left pedal opens valve IV and operates the plunger servo-motor. On releasing the pedals, valves III and V are opened by springs and the servo-motor pistons return to their initial positions.

The hydraulic pressure is 1700 lb./sq. in. and the net force available from the horizontal servo-motor is 12 tons and at the plunger 5½ tons. This is adequate

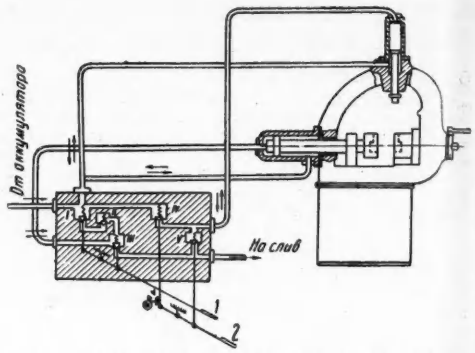

Fig. 6. **Diagrammatic arrangement of controls for press OVP-2.**
1—Right-foot pedal. 2—Left-foot pedal.

for bushes up to 2⅜ in. diameter with stamping pressures up to 2800 lb./sq. in. Both movements have a maximum stroke of approximately 8 in. The machine weighs 1 ton, uses about 1 gallon of pressure liquid per cycle and its speeds of operation are as follows:

plunger: 0·3 to 0·8 seconds "down" and 0·6 seconds "up";

matrix: closing 0·11 seconds, opening 0·3 seconds.

The Dies

A typical set of dies is shown in Fig. 7. It is best to design them so that only the internal surface is

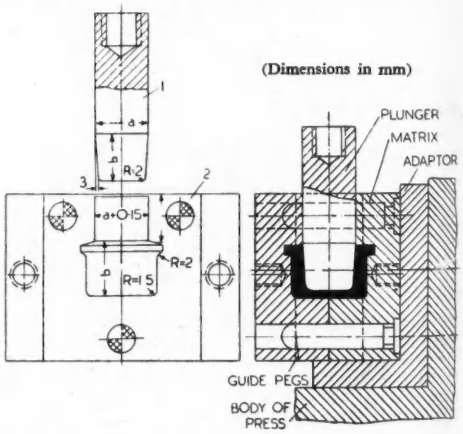

Fig. 7.

formed by the plunger (*cf.* Fig. 1, which represents the plunger as also forming the upper surface. This is a less successful design because there is nothing to prevent the stamping from lifting with the plunger and because a wrong quantity of metal poured into the mould will result in incorrect flange and/or bottom thickness, whilst with a correct design only the bottom would be affected and this is usually intended to be machined away later).

The dies must be made of low alloy heat-treated heat-resisting steel, with an allowance for shrinkage as follows: 40 per cent of normal unconstrained linear shrinkage of the alloy to be allowed on the diameters and 80 per cent on the height. Machining allowances are 0·030-0·040 in. radially on the inner and outer

diameters, 0·020 in. lengthwise at each end and 0 to 0·020 in. radially on the flanges. The taper required on the plunger is as tabulated above and should be extended about one-twelfth of an inch beyond the end of the stamped article to allow for fin formation. Average radii are as shown in Fig. 7.

Clearance between the plunger and matrix must be adequate to prevent seizure when the plunger heats up but it should be as small as possible to prevent splashing and fin formation. 0·006 to 0·008 in. on the diameter is usual. In this respect the length of the matrix neck which guides the plunger is also important: it must be so designed that the parallel part of the plunger is already in the matrix by about 2 in. before its bottom end touches the molten metal.

The matrix proper is kept as small as the size of the stamping will allow and L-shaped adaptors are therefore necessary for mounting them in the machine. These must overlap the joint at the bottom to support both halves of the mould. The fixed half of the mould must be easy to disengage from the adaptor to permit prompt remedial action in the event of plunger seizure.

The plunger should be lubricated and 5 per cent fine graphite in boiled oil is suitable, but all excess lubricant should be carefully wiped off.

Plant Design

The operating fluid can be drawn either from the general hydraulic pressure system of the works or from a special pumping installation in which case oil can be used instead of water. The capacity should be about 3 gallons per minute and the pressure 1700 to 2800 lb./sq. in.

If a specialized machine is not available it can be improvised, but it is essential to have power operation for the mould and plunger, and the normal speed of the latter must be not less than 1 ft./sec. with provision for increasing it up to 1·5 ft./sec.

The furnace must be so located that the operator can transfer the metal to the mould without moving his feet and without turning more than 90 degrees.

Gas for preheating the moulds, and air for cooling them must be readily available.

AUSTRIA

Two New Electrical Indicating Devices

By H. Hochrainer. (From *Elektrotechnik und Maschinenbau*, Vol. 64, No. 9/10, September/October, 1947, pp. 148-150, 3 illustrations.)

The following descriptions refer to two simple adaptor circuits which enable ordinary d.c. type moving-coil voltmeters to be used for the measurement of frequency and of the apparent and true watts of a circuit.

A. FREQUENCY-METER.

This adaptor device consists mainly of a current-regulator valve which compensates voltage fluctuations, a condenser giving a current dependent on frequency, and a resistor for the production of a reference current, as well as two dry-rectifier elements to rectify the two currents. The schematic diagram of the circuit is shown in Fig. 1. The current, before going to the condenser C and the resistor R, is kept constant by an iron-hydrogen current-regulator valve EW, so that fluctuations of the network voltage will not have an effect on the measurement current. The instantaneous vector sum of the two partial currents $\mathfrak{J}_C$ and $\mathfrak{J}_R$ will thus remain constant, even when frequency variations occur. By rectifying the potential drops of the two partial currents, two d.c. voltages u_C and u_R are obtained, which are equal when the capacitive current is equal to the ohmic current. Any variation in frequency will, therefore, cause also a variation of the partial current through the condenser, which will be inversely proportional to the change in frequency. As the total current is constant, owing to the iron-hydrogen resistance valve, the current through the ohmic resistance will vary in the contrary direction to the condenser current, that is, i_R will decrease when the capacitive current i_C increases, and conversely. Thus the voltage difference U_A between the terminals a and b will be twice as great as that due to the current variation through the condenser only, and the sensitivity of the circuit is correspondingly increased. Owing to the presence of the regulating valve, the fluctuations of the external a.c. voltage will not, within the limits determined by the valve, have any influence on the measurement, which is an advantage compared with other direct-reading frequency-measuring instruments. The sensitivity of the apparatus depends on the adjusted value of the total current and on the voltage available. The most suitable method is to transform to 220 volts when connected to a 110 volt supply, and to use the apparatus directly, without an intermediate transformer, when a network voltage of 220 volts is available. The standard regulator valves are generally designed for currents of 100, 150 or 200 milliamperes. The relationship between the measurement (d.c. output) voltage ΔU_A and the frequency f_0 is then given by

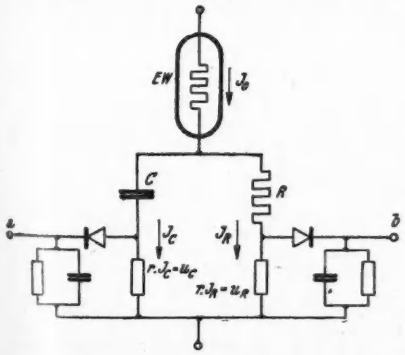

Fig. 1. **Schematic diagram of frequency-meter.**

$$\Delta U_A = \mathfrak{J}_0 r/(\sqrt{2} \cdot f_0) \text{ (volts per cycle per sec.)}$$

where $\mathfrak{J}_0$ is the adjusted value of the total current in amperes, r is the value (in ohms) of the auxiliary resistors used to obtain the potential drops for rectification, and f_0 is the frequency in cycles/sec. for which the frequency-indicator should give a zero reading. Taking for instance $\mathfrak{J}_0 = 0{\cdot}2$ effective amps., $r = 200$ ohms, and $f_0 = 50$ cycles/sec., we obtain:

$$U_A = 0{\cdot}2 \times 200/(\sqrt{2} \times 50) = 0{\cdot}56 \text{ volts/c.p.s.}$$

With a millivoltmeter, for instance, this circuit would have a reading accuracy of 0·1 per cent (as 0·1 per cent of 50 cycles/sec. = 0·05 cycles/sec.), which

corresponds to 28 millivolts. The measuring range can be extended to cover greater values without difficulty. The scale, however, will only remain linear as long as the percentage due to the frequency variation Δf can be neglected relative to 100, i.e. as long as $\sqrt{1 + (\Delta f/f_0)}$ is nearly equal to unity in the expression for the output voltage

$$U_A = \mathfrak{F}_0 \frac{r}{2} \frac{\Delta f/f_0}{\sqrt{1 + (\Delta f/f_0)}}.$$

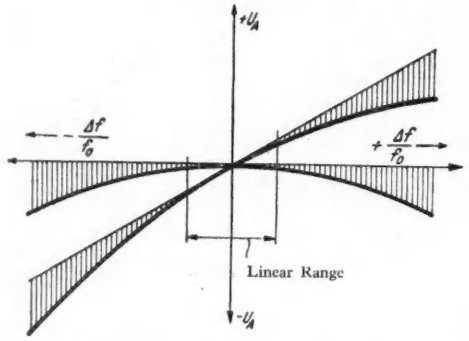

Fig. 2. **Frequency-meter characteristic.**

If this is no longer the case, then in accordance with the above equation, the curve shown in Fig. 2 is obtained which can be approximated by superimposing a parabola on the original straight-line characteristic, as can be seen by taking the first two terms of the binomial expansion of ΔU_A :

$$\Delta U_A \cong \mathfrak{F}_0 \frac{r}{\sqrt{2}} \frac{f}{f_0} \left(1 - \frac{1}{2} \frac{\Delta f}{f_0}\right).$$

This adaptor device for frequency measurements can be employed in conjunction with any moving-coil type voltmeter, and can also be used as a pick-up for remote-indicating equipment designed for direct current transmission.

B. WATT-METER.

For the measurement of the true or apparent watts output of a system one generally uses specially designed meters with a Ferraris disk, or indicators of the dynamometric type, which convert the power to be measured into a mechanical torque, with the pointer maintained in its equilibrium position by a spring force. The disadvantage of these instruments is that they require comparatively large moving masses and that it is impossible to transmit the results of the measurement over a great distance, unless the torque is converted into a corresponding a.c. frequency with various additional rotating masses and inertias in the system.

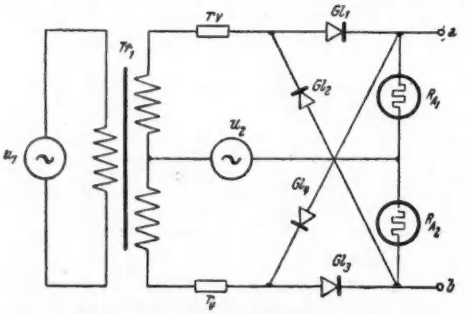

Fig. 3. **Schematic diagram of watt-meter.**

The new adaptor device, without resorting to moving mechanical elements or rotating systems, produces a direct currect proportional to true or apparent watts, and can be connected with a moving-coil type voltmeter to indicate the power output of a circuit, or it can be used for remote measurements, either directly or for the modulation of an oscillator unit.

A schematic diagram of the adaptor circuit is given in Fig. 3. It is a bridge-type circuit, in which two alternating voltages u_1 and u_2, proportional to input current and voltage, respectively, act simultaneously on a rectifier arrangement. This consists of 4 identical elements in series, which form a closed ring ; each of the two half-cycles of u_2 acts on two of the rectifier elements, in the same sense, so that the alternating current flowing from the voltage source u_1, which is proportional to the input current, will create on the two working resistors R_{A1} and R_{A2}, according to its phase φ, a direct-voltage difference which is primarily proportional to $\mathfrak{J} \cos \varphi$. As, furthermore, the working resistors $R_{A1,2}$ are hot conductors with a cold regulating resistance r_v in series, their resistance will also increase with increasing voltage, so that finally the difference of the voltage drops $\mathfrak{J} R_A \cos \varphi$ will be proportional to the power $U\mathfrak{J} \cos\varphi$. During the positive half-cycle, for instance, the rectifiers Gl_1 and Gl_3 are open, while Gl_2 and Gl_4 are closed. The regulating voltages themselves neutralize each other between the output terminals a and b. As functions of voltage, only the resistance values R_{A1} and R_{A2} remain. Passing through Gl_1 and Gl_3 the current flows during the half period through R_{A1} and R_{A2} and there, according to its phase angle relative to the regulating voltage, it generates a corresponding direct-voltage component. During the negative half-cycle, Gl_2 and Gl_4 are open, while Gl_1 and Gl_3 are closed. In the meantime the current has reversed its direction, so that any rectified component originating between a and b will have the same direction and the same magnitude as in the previous half-cycle, and the fluctuation of the output is reduced to a minimum.

As " hot conductors," ordinary 24-volt light bulbs with a consumption of about 2·5 watts are most suitable for this purpose. The output d.c. voltage indication is then about 1 volt/amp at 110 volts with $\cos \varphi = 1$. One must only make sure that the phase errors of the intermediate components are approximately equal, so that the resultant phase error of the measured quantity can be cancelled out.

The great advantage with this instrument, as in the case of the frequency-meter described above, is that it makes it possible to obtain measurements with any moving-coil type voltmeter ; the apparatus is simple to construct and enables remote measurements to be taken, and this is of considerable importance since the production of other types of remote-indicating instruments such as those using speed-compensation, impulse-type condensers, etc., is extremely difficult, quite apart from the mechanical shortcomings of these types of equipment.

Furthermore, the watt outputs measured in this way can be added to each other, by adding simply the corresponding d.c. voltages, that is, by making series connexions, provided an intermediate transformer is also used after the voltage transformer. An apparent-watts measurement is possible by inserting a special current transformer in parallel with a choke. This equipment using a minimum amount of material can be employed for various duties in modern electric plant testing and in running compound machines as well as in automatic control installations.

Measurement of Breaking Energy in Notched Rail Impact Test

By M. PERREY. (From *Revue de Métallurgie*, Vol. 43, No. 11/12, 1946, pp. 336-346, 10 illustrations.)

DURING the past fifteen years the technique of rail manufacture in France has made considerable progress and rails of steels of various qualities are now available. But the methods employed for the determination of the comparative impact strength of rails have been incomplete, cumbersome and inconclusive. It was recognized, however, that the solution to the problem must lie in the employment of an impact machine powerful enough to fracture the rail specimen with a single blow, such a machine to incorporate a device for measuring the breaking energy. A first step in this direction was an attempt to register the movement of the tup of the impact machine as a function of time. For this purpose a pencil was attached to the tup which inscribed a curve upon a chart wrapped around a cylinder rotating at uniform speed. This curve permitted to determine the variation in the speed of the falling tup before and after the impact and with it the breaking energy. However, the realization of this method met with considerable practical difficulties. Finally, in 1936 a machine was developed in which the movement of the tup was recorded by photographic registration of a beam of light reflected by a small mirror attached to the tup, the photographic film being wrapped around a cylinder revolving at constant speed. In order to obtain fracture at the proper point, two holes at a distance of 90 mm are drilled in the flange of the rail specimen and the flanges are then incised as indicated in Fig. 1.

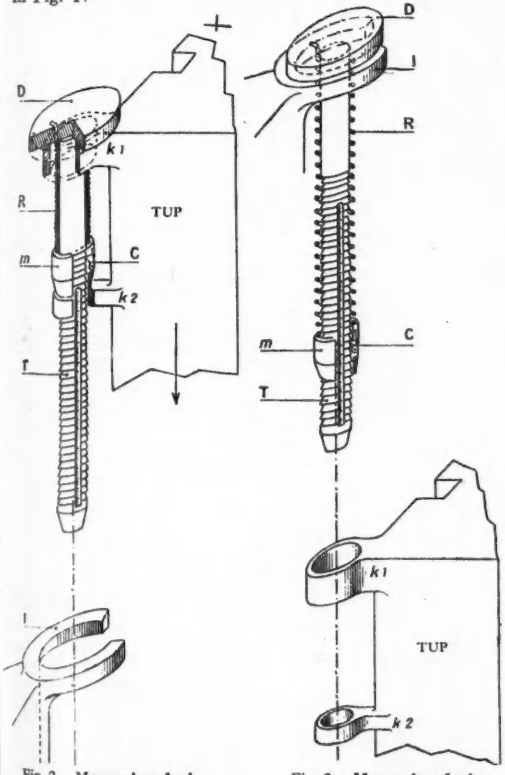

Fig. 2. **Measuring device prior to test.**

Fig. 3. **Measuring device after test.**

Fig. 1. **Test piece with incised flange.**

Subsequently the *Société des Hauts Fourneaux de la Chiers* at Longwy developed a patented testing device which is designed along totally different lines and entirely mechanical in action.

Principle of Operation. The kinetic energy of a tup of the weight P (kg.) which has fallen through a vertical distance of H (metres) is given by the product PH (kgm.). If, during its fall the tup encounters a piece of rail resting upon two supports and fractures it, then the tup will retain a certain residual kinetic energy, and the difference between this energy and the original kinetic energy PH represents the fracture energy augmented by incidental energy losses due to friction, vibrations, etc. Determination of the fracture energy can therefore be based upon measurement of the residual energy of the tup at a level conveniently situated below the point at which fracture of the rail occurs.

This measurement is carried out by attaching to the body of the tup a very small mass which, at the instant of measurement, will possess the same residual speed as the tup itself. This small mass consists of a threaded stem T which, at its upper end, is centrally affixed to the metallic disk D (Fig. 2). The latter forms an abutment for the helical spring R enveloping the stem T, and at its lower end the spring rests on the tailpiece or rider m ; the latter is provided with a pawl C which in turn engages with the threads of the stem. The manner in which the device is attached to the tup will be readily understood by reference to Fig. 2 which shows the stem resting in the brackets k_1 and k_2. As the tup descends further after having fractured the rail specimen, the stem T passes through the bore of the stationary stop I shown at the bottom of Fig. 2, whilst the head D cannot pass, with the result that the piece is lifted from its seats on the tup and thereby becomes seated on the stationary bracket I (Fig. 3). The sudden arrest of the piece causes the rider m, under the influence of its inherent kinetic energy, to slide downwards along the stem T until it comes to rest, the action of the pawl C preventing its return to its original position. The length of travel of the rider

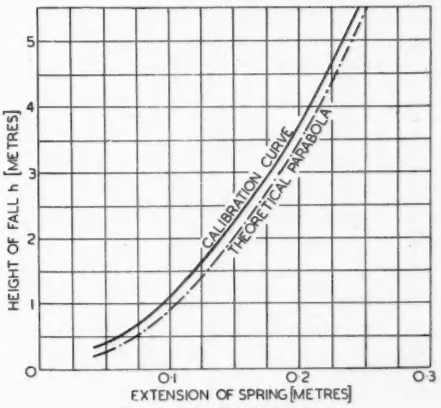

Fig. 4. **Calibration curve.**

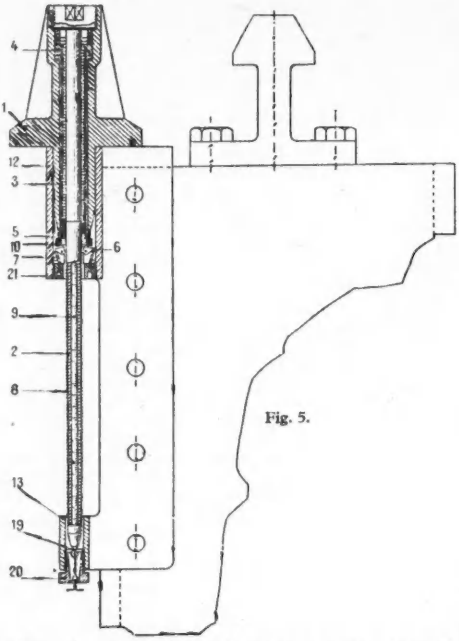

Fig. 5.

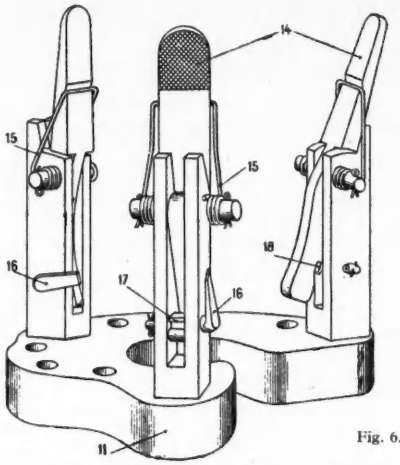

Fig. 6.

m along the stem T can then be seen on the scale provided in a groove as indicated in Fig. 3.

The device is calibrated by allowing the tup to drop freely from various heights and reading off the respective position of the rider on the stem T. By plotting the position of the rider versus the height of fall, a calibration curve is obtained which closely approaches a parabola, as will be seen from Fig. 4. By reading off the position x of the rider after fracture of a specimen, it is easy enough to ascertain the corresponding height h from the calibration chart, and the residual energy of the tup subsequent to breaking the test specimen will therefore equal Ph. The fracture energy will then be given by $T = PH - Ph = P(H - h)$ kgm., where H is the original elevation of the tup. The retardation of the tup due to fracturing the test specimen would cause the rider to shift and thus introduce an element of error ; but this is eliminated by a very simple mechanical device as described below.

The testing device as used in practice consists of three main parts, to wit :—

(*a*) The measuring device proper (Fig. 5) which comprises the head (1), the guide stem (2), the precision spring (3), the top windings of which are affixed to a ring (4) integral with the head, and a tailpiece or rider (5) attached to the bottom windings of the spring. The four pawls (6) are actuated by the rubber ring (7), so that they engage with the helical thread (8) of the stem (2). In this way the tail-piece or rider can des-

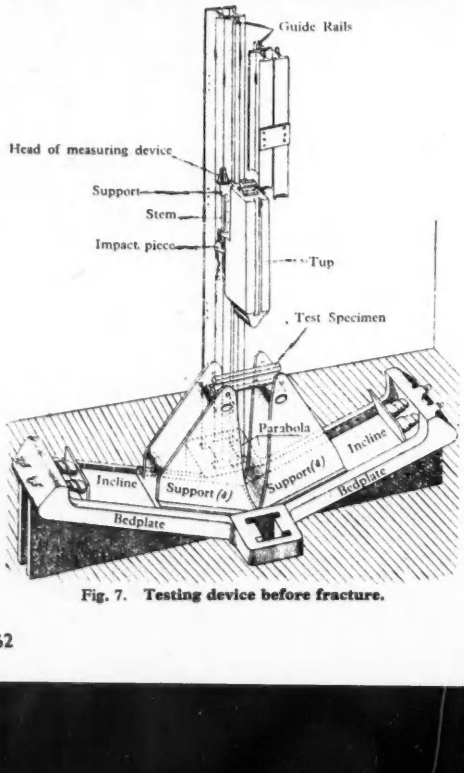

Fig. 7. **Testing device before fracture.**

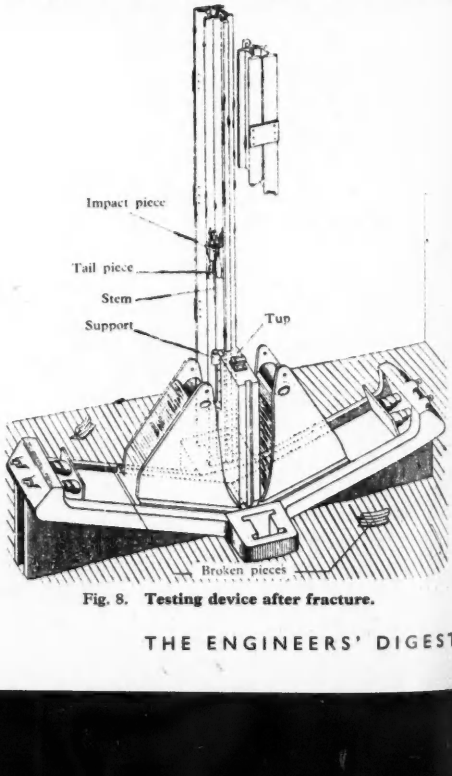

Fig. 8. **Testing device after fracture.**

cend along the stem but, by the action of the pawls, is kept from rising again. The position of the tail-piece is read off from the gradation (9) ; it can be removed by undoing the threaded ring (10). Fig. 5 shows the tup with the measuring device in position prior to the impact. The device is held down upon its seat on the tup by means of a copper wire extending from the hook (19) at the bottom end of the stem to the screw (20) affixed to the tup.

(*b*) The impact piece (11) which arrests the measuring device and lifts it off the tup is of horseshoe shape (Fig. 6) and is attached to a steel bracket which is bolted to the guide-rail column of the apparatus. This impact piece is provided with latches (14) operated by springs (15) and designed to prevent any rebounding of the measuring device after it has struck the stop (11). These latches are released by the measuring device striking the levers (16) of the releases (17). In operation, the impact of the measuring device upon the impact piece (11) causes the copper wire to be broken, thereby detaching the measuring device from the tup. At the same time the rider (5) moves downward because of its inertia and is then held in its lowest position by the pawls (6) engaging with the threads of the stem and preventing its return under the influence of the extended spring (3). Rebounding of the measuring device from the stop (11) is prevented by the action of the three latches (14), which upon their release, bear down upon the head (1) of the measuring device.

(*c*) Referring to Figs. 7 and 8, the two supports which carry the specimen to be fractured are slidably arranged on inclined tracks. The opposing sides of these supports are of parabolic shape. The tup entering the space between them subsequent to fracture of the specimen, is thus subjected to considerable braking action owing to the friction to be overcome and to the force required to separate the two sliding supports by wedge action and to push them up their respective incline. In this way smooth and comparatively shock-free action of the apparatus is achieved and tests can be carried out in rapid succession at the rate of some 30 to 40 fractures per hour.

SWITZERLAND

A New Moisture-Meter: The Diffusion Hygrometer

By H. GREINACHER. (From *Schweizerische TechnischeZeitschrift*, No. 13, March 27, 1947, pp. 189-194, 4 illustrations.)

THE most reliable method for measuring absolute humidity is to pass a known quantity of air through a tube containing a hygroscopic substance and to determine the weight of the water vapour absorbed. Another method is to connect the vessel containing the drying agent with the outside air, then to seal it off, and to measure the depression obtained which is equal to the partial pressure of the water vapour in the air. Thus, humidity can be expressed either in grammes per litre or in Tors (1 Tor = 1 mm. of mercury), and practically the same value is obtained with either method.

Three types of hygrometers are extensively used in practice, i.e. :—

1. The dew-point hygrometer ;
2. The wet and dry bulb psychrometer ;
3. The hair hygrometer.

In spite of the considerable amount of progress made in this field, it seems that there is still a need for a simple and reliable direct-reading hygrometer characterized by a high degree of stability, and based on a method which would be related as simply as possible to the measurement of vapour pressure. This has led to the development of an instrument based on the different rates of diffusion of water vapour and air through porous substances.

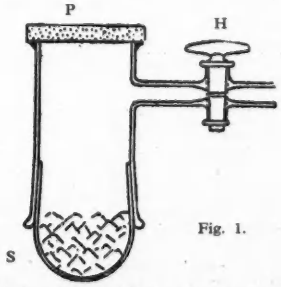

Fig. 1.

The principle of the diffusion hygrometer is illustrated in Fig. 1. A porcelain plate *P* of fine porous material is fixed with putty on to a glass tube which contains a drying agent placed in its removable lower portion *S*. The stopcock *H* is first opened to allow the internal and external pressures to be equal. When the stopcock *H* is closed, the pressures on both sides of the porcelain plates are equal, but whereas the internal pressure is that of dry air, the external pressure consists of the partial pressures of air and water vapour. The outside air pressure is then smaller than the internal pressure, hence the rate of diffusion of air from the inside outwards will be greater. Thus, on closing the stopcock, a negative pressure will develop, which will increase until it reaches a value at which the rates of diffusion of air in both directions will be equal. The water vapour also diffusing into the vessel is absorbed by the drying agent, and only has an indirect effect on the internal pressure, by acting against the diffusion of air from the inside outwards. As it is hardly possible to develop semi-permeable plates which would be non-permeable to water vapour, the negative pressure will be slightly lower ; but the difference only amounts to about 10 per cent of the partial pressure.

The diffusion hygrometer is based on the assumption that the negative pressure Δp_t can be regarded as proportional to the pressure p of the water vapour. This relationship has been confirmed by tests and has also been explained theoretically (see *Helvetica Physica Acta*, 1944, Vol. 17, p. 437). We can thus write :

$$p = k\Delta p_t \quad .. \quad .. \quad .. \quad (1)$$

where k is an instrument constant. This constant can be determined by placing the vessel (Fig. 1) under a bell jar with a bowl of water, that is, in 100 per cent humidity, and reading first Δp_t and then taking p for the corresponding temperature from a table for vapour pressures. When p is obtained by means of formula (1), the relative humidity $f = 100\ p/p_s$ is also known, since the saturation pressure p_s can be determined from psychrometric tables.

The instrument constant k can be eliminated by using a double instrument, with two vessels, the second vessel containing distilled water. On closing the stopcock *H* there will be a positive pressure Δp_t, proportional to the difference between the partial pressures of water vapour inside and outside the vessel, that is, proportional to the saturation deficit $p_s - p$. Then

$$\Delta p_t = k\,(p_s - p) \quad .. \quad .. \quad (2)$$

The relative humidity is then obtained as a percentage from (1) and (2) :

$$100\,\frac{p}{p_s} = 100\,\frac{\Delta p_t}{\Delta p_t + \Delta p_t}, \quad .. \quad .. \quad (3)$$

and the saturation deficit is

$$100\,\frac{p_s - p}{p} = 100\,\frac{\Delta p_t}{\Delta p_t + \Delta p_t} \quad .. \quad .. \quad (4)$$

Thus neither a temperature reading nor a table is required to obtain relative humidity, but two pressure readings will always be necessary. The two manometers required can be graduated for any units of pressure.

In the first simple model of the diffusion hygrometer* each of the glass vessels is connected to a liquid manometer. A stopcock is not needed, as the diffusion pressures which build up are always equal and independent of the initial state of the apparatus. When the glass containers are first inserted the pressures are of course not equal, because one vessel contains water, and the other a drying agent. Paraffin was used as a manometer fluid, as it has a low specific weight and does not evaporate, while KOH rods were used as drying agents. The porcelain plates on each element were about 5 mm. thick and had an area of about 2 cm². The scale was graduated in millimetres. To eliminate calculations according to formulæ (3) and (4), a table can be used which gives f directly for each value of Δp_t and Δp_t.

In improved version of the hygrometer* the two vessels are in a metal box closed by a mirror-glass plate on the front side. The two elements with the porcelain plates protrude through the back wall, and are bathed freely by air without being disturbed by the observer. To avoid having to take two readings from each manometer to determine pressure differences, the scale is arranged so that it can be shifted upwards or downwards, by means of a screw-head until a fine horizontal wire at the height of the zero-line comes to the level of the outer meniscus of the U-tube. On the inner branch the pressure difference is then read directly on the scale. A small metal thermometer (not required in principle) completes the equipment.

Although the apparatus is simple and easy to handle, it has the drawback of having liquid manometers which make it delicate to transport. It is an advantage to use a metal manometer with a pointer and dial instead of the liquid manometer, and a metal manometer was therefore developed, consisting of two fine pressure elements, which were both provided with scales graduated in millimetres of water, one measuring negative and the other positive pressures, the ranges being from 0 to 20 and 0 to 30 mm of water, respectively. Such manometers are rather expensive, however, so that they would not be used extensively in practice. If a single-element instrument is employed it immediately involves the use of tables of vapour pressure and an instrument constant.

A single-element instrument* was developed, however, which gives direct readings without using tables or formulæ. This was done by designing the dial of the manometer so that it gives directly the relative humidity. An outer graduated circle is the scale for the diffusion pressure, which serves first to obtain the absolute humidity, and smaller concentric circles indicate the degree of moisture from 0 to 100 per cent.

The apparatus is very easy to handle ; it does not use a drying agent but uses distilled water introduced through a small metal tube on the back side. After unscrewing the cap a small glass tube filled with water is pushed in from below, and after screwing the cap on again, the apparatus is in working condition and does not need refilling for about a month. The thermal diffusion process begins immediately, and after a short period of time the stationary diffusion pressure is reached.

The calculation of the isothermals is somewhat more tedious with water used instead of a drying agent. The instrument constant k is first determined as indicated above and, after finding the saturation pressure of the water vapour for a given temperature, one obtains for the corresponding diffusion pressure the relationship $\Delta p_s = kp_s$. Furthermore the diffusion pressure for any degree of relative humidity f is given by

$$\Delta p = k\,(p_s - p) \text{ or } \Delta p = \frac{\Delta p_s}{p_s}\,(p_s - p),$$

and as by definition $f = 100\,p/p_s$, we have

$$\Delta p = \Delta p_s\,(1 - f/100),$$

$$\text{or} \quad f = 100\left(1 - \frac{\Delta p}{\Delta p_s}\right).$$

In this way one can obtain for each Δp, that is, for each position of the pointer, the corresponding humidity f. If we now mark the corresponding points for a series of Δp, the line drawn through these points will be the isothermal.

Another type of direct indicating instrument can be made using a pointer and dial type thermometer, which eliminates the initial temperature reading required with the hygrometer. The pointers of the manometer and of the thermometer are moved until they intersect each other. The point of intersection then corresponds to a definite degree of humidity. The curves of equal humidity can thus be plotted on a nomogram, progressing for example in steps of 5 per cent at a time. The curves, of course, are no longer circles, and the relative humidity can thus be read at the intersection of the two pointers. The circular arcs described by the ends of the two pointers can also be graduated so as to indicate the diffusion pressure (that is, absolute humidity) and temperature. A model of this type is now being built.

The diffusion hygrometer will give reliable results even after many years of use, and it is not affected by dust and other particles in the air. The rapidity with which the diffusion hygrometer follows humidity variations depends on the size, thickness and quality of the porous plate. The larger the plate the more rapid will be the response, but it is not advisable to choose too large a size for the plate as the water or the drying agent will be used up too rapidly and therefore require frequent replacement. Quick response is also improved by small size vessels. Using small container volumes and a porcelain plate of 2 cm², it was necessary to refill only once every three months. However, the instrument takes about 10 minutes to settle down after a considerable variation in humidity. The model last referred to has a larger volume because of the manometer, and uses a plate with an area of 24 sq. cm. The water introduced in one filling is sufficient for a month and can easily be renewed.

The accuracy of the measurements increases with temperature. At low temperatures the measurement is less accurate because the partial pressures and hence the diffusion pressures are low. To obtain good results it is necessary to maintain the room or apparatus at a constant temperature for some time before taking the readings. Any rapid temperature variation (for instance in the vicinity of heaters or radiating bodies) is the cause of a pressure variation in the apparatus, which can only be compensated by diffusion. Provided this condition is fulfilled, the instrument will work automatically and give reliable results.

*Illustrated in the original article.—Editor, E.D.

CLASSIFIED ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns, are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

BEARINGS AND BEARING METALS

Bearing Construction with Cast Iron as Anti-Friction Material.

By M. S. KARPYSHEV. (From *Vestnik Machinostroenia, Russia*, Vol. 16, No. 11-12, 1946, pp. 24-29, 6 illustrations.)

THERE is a widespread belief that cast iron can only be used as an antifriction material under pressures up to 20 kg/cm², at rubbing speeds less than 2 m/sec., and with steady shock-free conditions. It has been seen, however, that it is not essential to keep the values of p and v to such limits if the machining is precise and produces good finishes and if design and lubrication are satisfactory. The author has carried out a series of tests and is of the opinion that cast iron is suitable for use in bearings particularly for rocking or reciprocating parts.

Careful broaching of bearings made them able to withstand high specific pressures without scuffing, and it was found that when p is high and v is less than 0·1 m/sec. full length lubricating grooves should be used, but this is not permissible when v is high and other improved lubrication arrangements are necessary whilst the bearing brushes must be relieved from bending stresses.

Scuffing and scoring will occur if machining is inaccurate, surfaces are not smooth and clean and subject to bending stresses, whilst even more exacting conditions are necessary when the shaft has a hardness lower than 45-55 R_c.

The tests were carried out on bearings of an Alligator shears jaws, and these shears were used under normal conditions and observed for three years. They were frequently subjected to overloads of 150 to 200 per cent over rated values, but there was no deterioration of the bearings. The values of p and v greatly exceed those normally recommended.

The ratio l/d of journals should be as low as possible and never more than unity. The lower it is the higher the values of p and pv can be. Cast iron bushes are more susceptible to stepped load application than bronze brushes, because under such conditions the distribution of pressure over the length of the bearing is not uniform.

BLAST FURNACE PRACTICE

Research into the Causes of Iron Ore Splitting in the Blast Furnace.

By M. J. FORTADO. (From *Revue de Métallurgie, France*, Vol. 43, No. 7-8, July/August, 1946, pp. 219-228, 2 illustrations.)

BLAST furnace operation is improved by previous crushing and sizing of the ore. This also reduces the amount of dust carried by furnace gases since the fines are eliminated by sieving, and ore splitting in the furnace is prevented or much reduced. Ore splitting has been observed in blast furnace practice and found to be due to (a) moisture separation at about 300 deg. C., and (b) de-oxidization and de-carbonisation at about 600 deg. C. In laboratory tests and by blast furnace observation, the author has investigated the mechanism of these phenomena. Moisture separation between 100 and 400 deg. C. leads to internal tension in the particles and subsequent splitting according to their size, the nature of the ore, and the rate of heating. Slow heating or drying before heating prevents splitting. De-oxidization in the reducing furnace gas atmosphere begins at 400 deg. C., and de-carbonisation at 550 deg. C., with both reactions starting at the ore surface and slowly penetrating into the pieces. Internal stresses cause the bursting of the particles and the generation of dust. Bursting was not observed during heating in an oxidizing atmosphere.

Particles smaller than 2 inches across do not split, but bigger particles may burst, and, other factors being equal, a maximum size which just does not split can be determined for every particular kind of ore. It is found that the lime content of ore reduces the tendency to splitting, and ores can be graded for suitability according to their combined content of iron and lime. Less suitable ores must be crushed to a finer size if splitting and excessive dust are to be avoided. A comparison of dust carried by blast furnace gases with dust from heating and splitting tests shows that although the greatest part of the coarser dust due to splitting remains in the furnace, the finer dust of lower iron content can be identified in the blast furnace gas.

BOLTS AND NUTS

Engineering Aspects of Bolt Variables.

By V. E. HILLMAN. (From *The Iron Age, U.S.A.*, Vol. 160, No. 21, November 20, 1947, pp. 62-68, 5 illustrations.)

AN important, but too often ignored, phase of metal product manufacture is the efficiency obtained from bolted assemblies. This article discusses, from a practical engineering viewpoint, the function and limitations of a bolt in a tapped hole or nut and offers many helpful ideas for obtaining better results from the common bolted assembly. Procedures for determining the tension induced in a bolt during tightening are set forth, and such variables as machining, heat treatment, and steel analysis are explored. A table on defects in screw threads, their causes and defects, is also presented.

Summarizing, it may be said that ways and means of determining accurate bolt tightness are impracticable in most machine assemblies. The micrometer method of tightening is sufficiently precise but not always practical for any present engineering need.

Excessive tightening encourages stretch rather than security.

A bolt stressed too much beyond its yield strength may not adequately serve as a fastener.

The degree of tightness in a bolted assembly should be a function of its working load.

For most practical purposes it is desirable to stretch a bolt so that its tension will not be more than 80 per cent of its yield strength.

The tension induced in a bolt during tightening may be measured by checking its elastic elongation.

The torque wrench is the most popular device for specifying nut tightness.

The angular turn method of measuring nut tightness is still to be perfected.

Tables specifying nut tightness should be specially developed for each assembly line. The terms in which nut tightness is specified may be foot-pounds, elastic elongation per inch of bolt length, or the degrees of angular turn.

Wear and abrasion of the abutments will develop bolt looseness.

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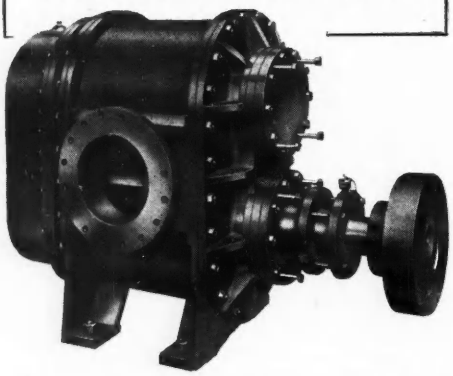

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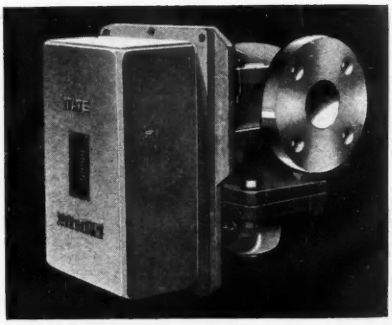

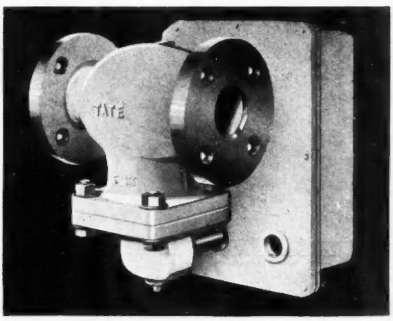

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Bolts and Nuts—*continued.*

Important considerations that enter into the proper functioning of a bolt are : (1) the yield strength of the material ; (2) the tension developed in tightening ; (3) the duty imposed in service ; (4) the elasticity of the abutment ; (5) loss of tightness in service by wear ; and (6) abrasion and corrosion.

Bolt looseness may be the result of improper tightening procedure.

Bolt steels, heat treatments and physical properties are adequately standardized.

A steel of minimum yield strength should be selected consistent with operating and service requirements.

ELECTRIC BATTERIES

Self Discharge in Lead-Acid Storage Batteries.

By A. C. ZACHLIN. (From a paper presented to *The Electro-chemical Society, U.S.A.*, Preprint 92-28, October, 1947, pp. 323-343, 11 illustrations.)

THE effect of proportion of antimony in grid metal on the rate of self discharge of lead-acid storage batteries was investigated over the range 0·5 per cent to 12 per cent Sb with the conventional type of cells, and also with the low discharge type of construction. As the proportion of antimony in the grid metal increases, the rate of self discharge increases. This rate is higher and accelerates upward more rapidly at 125 deg. F. than at 80 deg. F. To produce a cell really free from self discharge the use of pure lead metal for grids is required. Alloys with 1 per cent to 2 per cent Sb are difficult to cast into grids and therefore are not practical. Tellurium, where added to pure lead grid metal to the extent of 0·06 per cent, does not markedly affect the self discharge of cells and does not contribute appreciably to the mechanical properties of the grid. Negative plates lose power on standing from 2 to 5 times faster than do the positive plates. As the concentration of electrolyte increases from 1·250 sp. gr. to 1·400 the rate of self discharge increases slowly for cells with pure lead grids and rapidly for those with antimonial lead grids. As the length of test on open circuit becomes longer the rate of self discharge decreases somewhat. Low discharge cells with pure lead grids were found to lose capacity so slowly at 80 deg. F. that after a 35 month period of standing on open circuit they still had half of the original capacity left available. At 125 deg. F. they lose capacity much more rapidly, but even then not materially exceeding the rate for the conventional batteries at 80 deg. F. The material of separators, comparing treated wood with rubber, does not affect self discharge. Cell design, however, as it depends on the design of the grids, does affect the self discharge behaviour of antimony.

ELECTRIC INSTRUMENTS

Apparatus for the Generation of Two Electrical Pulses Separated by a Given Time Interval.

By K. DEUTSCH. (From *Elektrotechnik und Machinenbau, Austria*, Vol. 64, No. 11/12, November/December, 1947, p. 182, 2 illustrations.)

FOR various purposes it is necessary to have an apparatus which can generate two electrical pulses separated by a given time interval. The Helmholtz pendulum is an apparatus of this type but it has the disadvantage of being very heavy, and, because of its high sensitivity, it is more of a laboratory type of instrument.

A similar apparatus fulfilling the requirements has been developed which is more mechanically robust and less sluggish in operation. It consists of a circuit comprising two piezoelectric elements. A steel ball K_2 is held by an electromagnet M_2 at the top of tube 2 (see Fig. 1). When the current in the magnet circuit

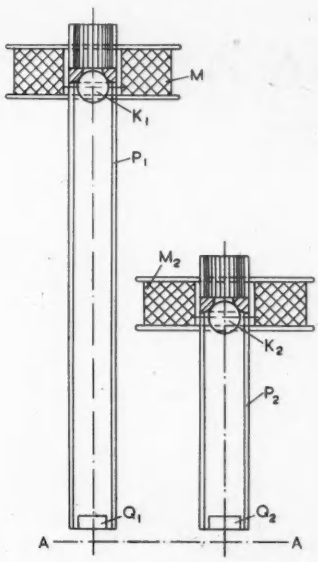


Fig. 1

is cut off the ball is released and impinges upon a quartz disk Q_2 after a time $t_2 = \sqrt{2s_2/g}$. The quartz, owing to its piezoelectric properties, generates under the pressure of the impact a voltage pulse E_2. The electromagnet M_1 is cut off at the same time as M_2, but, as its tube is longer, the ball K_1 will reach the quartz disk Q_1 after a time $t_1 = \sqrt{2s_1/g}$, and a second voltage pulse E_1 is then generated. The time between the two pulses is $t = t_1 - t_2 = \sqrt{\frac{2}{g}}(\sqrt{s_1} - \sqrt{s_2})$. The correct value of g for the corresponding latitude should be used, as when using the Helmholtz pendulum. The quartz disks can be moved up or down within their tubes, to give the time interval required. For example, when $s_1 = 19{\cdot}62$ cm and $s_2 = 11{\cdot}04$ cm, the time interval $t = 50$ milliseconds. To bring the balls back to their original position the cylinders are tilted over so that the spheres can roll back to their respective coils. The current is then switched on and the apparatus is set upright again for the next test.

The tubes P_1 and P_2 must be of a material with a very low thermal coefficient of expansion so that the distances s_1 and s_2 may remain as constant as possible. The electromagnets must release the two balls simultaneously and this is achieved by means of a special circuit (see Fig. 2). When the switch S is opened, the current drops exponentially from an initial value i_0. Let $i_{1,2}$ = the currents through coils $M_{1,2}$ for

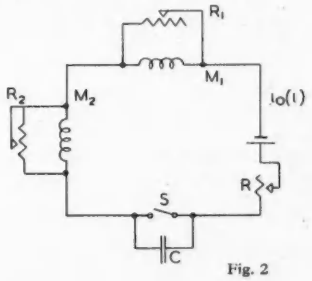
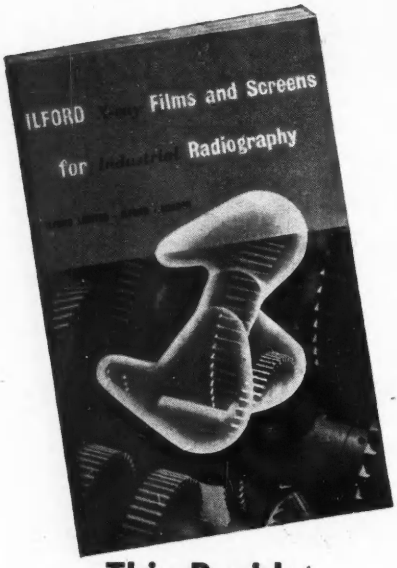


Fig. 2



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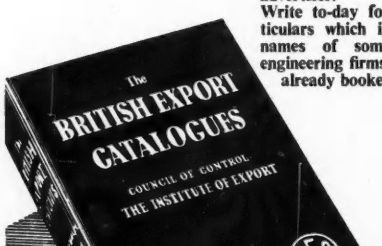

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which the spheres $K_{1,2}$ will just begin to fall ; i is a certain value of the total current which is less than the initial value i_0 ; and $i_1', _2$ are the currents in the resistors $R_{1,2}$ when the total current is i.

Then by adjusting R_1 and R_2 so that $i_1 + i_1' = i_2 + i_2' = i$, the two balls will start falling at the same instant, when $i_{total} = i$. There will, however, still be small fluctuations Δi between the instants at which the two spheres begin to fall, which cannot be eliminated by R_1 and R_2. The accuracy of the apparatus is given by

$$\Delta t = \Delta t_s \frac{\Delta i}{i_0}$$

where Δt_s is the switching-off time.

HEAT TREATMENT

Tensile Strength of Case-Hardened Layers.

By A. M. PALY. (From *Zavodskaya Laboratoria, Russia*, No. 11-12, 1946, pp. 987-988.)

THE hardness of case-hardened surfaces is a measure of resistance to plastic deformation only and, therefore, it is no criterion of resistance to fracture. Yet this latter quality, as measured by a tensile or bending stress, is of great importance in a hardened layer. We see evidence of this in hair-line cracks which develop during grinding, in the crumbling away of the hardened surfaces of gear teeth, and in the failure of cutting edges in taps, etc.

It is particularly important to know how the hardened material is affected by low temperature tempering which does not appreciably affect hardness but is intended to remove some of the internal stresses. Reference books tend to give no guidance as to the time which is required for this operation and common practice is ½ to 1½ hrs. The only way to determine whether this is sufficient is by tests designed to produce failure in tension, i.e., either tensile or bending tests to destruction.

The author chose bending tests as being more reliable and conducted tests in an Amsler machine on specimens of 0·8 × 0·4 in. steel, carburized throughout and given their appropriate hardening treatment, followed by varying periods of tempering at 160 deg. C. in air.

Average improvement in tensile strength was 32 per cent with 1 hour of tempering and 51 per cent with 2 hours and 3 hours. Beyond that the improvement, if any, was small. The Rockwell hardness fell from 64-65 in untempered specimens to 61-63 after 3 hours and remained constant after that.

Experiments on the same lines with hardened low-alloy steel gave the same results and the author also found that cyanided gear surfaces which were liable to cracks responded perfectly to low temperature annealing. His conclusions are that all tetragonal martensitic structures require 2-3 hours of low temperature annealing and that it was wrong to use hardness tests alone when comparing the merits of various forms of heat treatment ; they should always be supplemented by tensile tests.

MACHINE TOOLS

Development of a High-Speed Lathe.

By R. L. TEMPLIN. (From *Machine Design, U.S.A.*, Vol. 20, No. 1, January, 1948, pp. 140-142, 3 illustrations.)

PRACTICAL limitations on the maximum machining speeds for aluminium so far have been imposed by the limitations of the machine tools rather than by the machining properties of aluminium.

After consideration of the many factors involved, it was decided to develop an experimental turret-type lathe with a spindle having an internal bore of about 2½ inches which would operate at a maximum speed between 7,500 and 10,000 r.p.m. Two serious problems were immediately evident : What kind of bearings and what type of drive should be used for the machine ? Consideration was given to ball, roller, and plain bearings before deciding to use the Filmatic segmental type.

For controlling the temperature of the lubricating oil used for the spindle and motor bearings, a 5 h.p. hermetically sealed "Air-Temp" cooling unit was used.

The high speed of the spindle with a direct-connected motor indicated a squirrel-cage type motor which has a rating of 80 h.p. at 220 volts, 3 phase, 60 cycle, 3,450 r.p.m. This same horse-power rating is maintained as the frequency is increased to 150 cycles, which corresponds to a top speed of about 8,800 r.p.m., and during this frequency and speed increase the voltage is raised to a maximum value of 350 volts. The stator is provided with water cooling.

The motor was connected to the back end of the spindle by means of a dynamically balanced, forged aluminium alloy Thomas flexible coupling.

A hydraulically operated tool carriage with a separate hydraulic circuit was provided, and a manually operated cross-slide and a turret type tool holder were mounted on the tool carriage. A heavy welded steel bed was provided for the machine on which the motor, spindle housing, and tool carriage assembly were mounted.

A collet chuck of 2¼ in. maximum capacity is operated manually by a lever mounted on top of the spindle housing. In addition, a special 10 in. three-jaw scroll type chuck was furnished.

From the results of tests conducted on this experimental lathe, a number of conclusions appear to be warranted, of which the following are typical :

1. Aluminium alloys can be satisfactorily machined by turning at cutting speeds up to 20,000 f.p.m. No indication of an upper speed limit was observed.
2. The number of cubic inches of metal which can be removed per minute for each horse-power going into the cut increases with increasing rate of tool feed. In these tests as much as 7·5 cu. in. of metal were removed for each horse-power developed at the cut.
3. The net horse-power required to remove a given volume of metal is independent of cutting speed and depth of cut.
4. As much as 470 cu. in. of metal per minute (approximately 47 lb.) were removed without overtaxing the equipment. There was no indication that greater amounts could not be removed.
5. The amount of power required to drive the spindle when a cut was not being made was greater than had been anticipated. Extrapolation of the data indicate that the 80 h.p. motor might be operating at as much as 70 per cent of capacity when driving the idle spindle at 9,000 r.p.m.
6. Changing the oil in the lubricating and hydraulic system to a lighter oil resulted in an average reduction in idle horse-power of approximately 17 per cent.

METALS CORROSION

Metal Surface Protection by Tin Diffusion.

COMMUNICATION FROM THE GERMAN COPPER INSTITUTE, BERLIN. (From *Metalloberfläche, Germany*, Vol. 1, No. 8, August, 1947, pp. 185-187, 6 illustrations.)

HOT tin dipping has the disadvantage of a noticeable dimensional increase of the coated detail and thus a high tin consumption, whilst electrolytic tinning has to be controlled within fine limits, if a reliable deposit is to be obtained, and presents considerable difficulties for parts of complicated shape. The diffusion process

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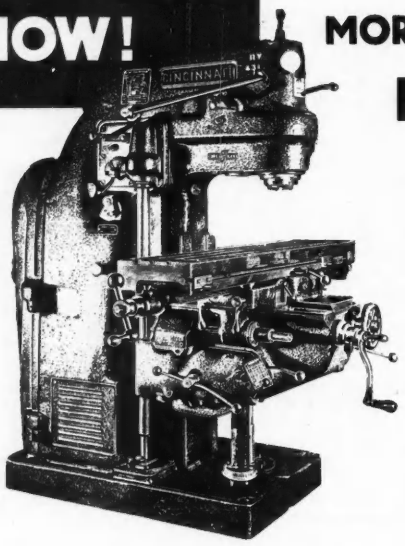

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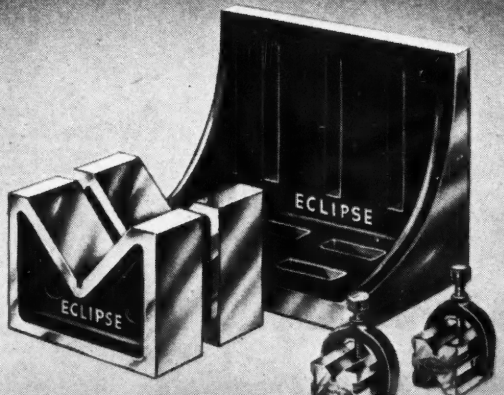

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gives reliable, thin and uniform surface protection, however, without noticeable dimensional increase. A salt bath containing $SnCl_2$ is held at 350 to 450 deg. C., and the part is dipped into it. Surface atoms of the metal to be protected combine with chlorine and allow the freed tin to diffuse into the metal surface and replace the metal halogenides, according to the formula $Me + SnCl_2 \rightarrow MeCl_2 + Sn$. The process can be applied to copper, cuprous alloys, and steel. On brass, a satisfactory protecting layer of 0·002 inches is formed in about 15 minutes, with a consumption of 0·23 oz. of tin per sq. foot. As tin atoms partly replace metal atoms, the increase in thickness due to the protecting layer is less than 0·0001 inches. On steel, a layer of 0·00015 inches is formed in 15 to 60 seconds, the dimensional increase being 0·00006 inches and the tin consumption 0·1 oz. per sq. ft.

The equipment consists of a heated graphite crucible, a pyrometer, and a salt bath with technically pure potassium, sodium, zinc chlorides and tin salts, i.e. complex tin-chloride salts or, in a modified process, pure tin. The metal parts are de-greased, de-oxidized, and then dipped into the bath for the correct time, taken out, washed under water, and dried. The relatively low bath temperature prevents distortion, reduces gas consumption and wear of the crucible. The salts are inexpensive, the bath can easily be regenerated, tin consumption and losses (the bath contains 1 to 2 per cent tin) are very small.

Investigations in German shipyards have proved the good corrosion resistance of the protective layer and its applicability to high pressure work. If surface wear is to be expected, the thickness of the layer should be increased.

STEAM POWER MACHINERY

All-Union Conference on Steam Turbines.

(From *Kotloturbostroenie, Russia*, No. 3, 1947, pp. 28-29.)

In March, 1947, an All-Union Conference on Steam Turbine Technique was held at Leningrad. Reference was made to the 100,000 kW steam turbine running at 3,000 r.p.m. with initial steam conditions of 90 atmospheres at 480 deg. C. built by the Leningrad Works "Stalin." Future work includes the development of topping turbines for steam conditions of 175 atm. and 550-600 deg. C. The Economiser Works are reported to be engaged on the design of high pressure turbines for centrifugal boiler feeders with initial steam conditions of 90 atm. at 480 deg. C. and a back pressure of 1·2-2·5 atm. Turbine capacity is 1,350 kW and its efficiency at the coupling is 70 per cent.

The report also states that at the Central Boiler and Turbine Institute a new aerodynamic test plant equipped with optical instruments of the Toepler type was installed. A large aerodynamic wind tunnel plant is being erected at that institute which will include an air compressor capable of delivering 30,000 cu. m. of air per hour at 7 atm. pressure. Methods for the computation of turbine blade profiles are being developed and investigations into erosion and corrosion in the last turbine stages are being made.

At the same institute important work on gas turbines is in progress and the characteristics of both open-cycle and closed-cycle types have been studied. A single stage axial compressor was installed in 1946 and a first series of tests was run. Experiments aiming at the development of a three stage axial compressor are under way and an experimental single stage gas turbine is being built. At the laboratory of the institute the fatigue strength and the creep properties of turbine materials at 800 deg. C. are also being investigated.

The preliminary layouts of two mercury-steam plants were completed in 1946. The first of these projects is a 4,000 kW mercury turbine to form part of a 10,000 kW mercury-steam plant. This is to be installed at the Central Boiler and Turbine Institute. Mercury vapour will be generated at 10 atm. and 515 deg. C., and the overall plant efficiency is estimated at 40 per cent. The second mercury-steam plant is intended for the driving of blast furnace blowers, and the exhaust mercury vapour is to be used for heating the blast.

VIBRATIONS

Permissible Torsional Vibrations of Piston Engines.

By A. Kleiner. (From *Schweizerische Bauzeitung, Switzerland*, Vol. 65, No. 33, August 16, 1946, pp. 443-448, 6 illustrations.)

Torsional vibrations of piston engines are often calculated under the assumption that the crank and transmission shafts are rigid. Speed variations and the amplitudes of angular vibration thus calculated are smaller than those actually occurring, and this error is compensated for by exaggerated limitation of these values in acceptance tests of prime movers. This leads to the design of unnecessarily large flywheels. As a matter of principle, only the final use of the prime mover should determine acceptance conditions, e.g. the limitation of voltage variation for flicker-free lighting or the limitation, in electrical degrees, of the vibration amplitude for parallel running of a.c. generators. The required coefficient of speed variation can then be calculated and may have different values according to cylinder number and r.p.m. In order to comply with limitations thus imposed, the speed variations and angular vibration amplitudes of the prime mover must be determined accurately and the torsional elasticity of the shaft must not be neglected. A system of equations is then obtained for the various harmonics of speed vibrations, and these harmonics have to be added in different ways according to amplitude, frequency and phase. Any dangerous frequency can thus be specially determined and limited without unnecessary increase of the general level of safety. With modern Diesel engines, however, one single harmonic of an order equal to half the number of cylinders for four-stroke engines is more pronounced, the others can be neglected except for special frequency conditions.

The general solution of the vibration equations shows damped natural and, more important, undamped forced vibrations. The damping factor of the flywheel can be neglected except near resonance. In the case of resonance, the resulting coefficient of speed variation refers to the engine plus a particular flywheel and changes when the flywheel is changed. For "homogeneous" arrangement of the engine (equal inertias and shaft torsions), the angular vibration amplitude can be calculated as a combination of a homogeneous function (engine) and an inhomogeneous function (flywheel). Nomograms help to shorten calculations in this case. The arrangement of a generator rotor connected by a flexible shaft to the flywheel can be theoretically reduced to the simpler case of a single flywheel.

Calculation under the assumption of a rigid shaft is sufficiently correct for low speed single cylinder engines but is the more inaccurate the nearer the exciter frequency is to the natural torsional frequency of the shaft. For modern engines, the approximation is often so inaccurate as to be valueless. However, torsional vibrations of modern piston engines are nearly always calculated at the design stage. The basic data for exact calculation of torsional vibration amplitudes and speed variations are thus given and the remaining calculations are simple. Acceptance tests should therefore be based on exact calculations and requirements.

NEWS OF THE MONTH

Announcements in this section include News relating to British Firms, and on Equipment produced by British Manufacturers.
Available literature may be secured by addressing a request to the advertising department of "The Engineers' Digest," or by writing direct to the manufacturer and mentioning "The Engineers' Digest" as a source.

- PERSONAL
- NEW EQUIPMENT
- BOOKS AND CATALOGUES
- BUSINESS CHANGES
- NOTES

PERSONAL.

Mr. H. Boyd Brown, B.Sc. (Eng.) has been appointed sales manager of the motor department, and **Mr. C. H. De Nordwall, M.I.E.E.,** sales manager of the industrial control department of Metropolitan-Vickers Electrical Co., Ltd. **Mr. E. A. Guthrie** has been made manager of the American Division, and **Mr. F. R. Mason, A.M.I.Mech.E.,** principal representative of the Metropolitan-Vickers Electrical Export Company, Ltd.

Mr. R. C. Bond has been appointed chief officer (locomotive construction and maintenance); **Mr. E. Pugson,** chief officer (carriage and wagon construction and maintenance); **Mr. C. M. Cock,** chief electrical engineer; **Mr. A. Dean,** chief officer engineering (works); **Mr. S. E. Parkhouse** and **Mr. E. W. Rostern,** chief officers (operating); **Mr. H. Rudgard,** chief officer (motive power); **Mr. O. H. Corble,** chief officer (marine); **Mr. H. A. Short,** chief officer (docks); **Mr. J. Ness,** chief officer (new works); **Mr. E. S. Cox,** executive officer (design); **Mr. A. E. C. Dent,** executive officer (road motor engineering); **Mr. H. H. Dyer,** executive officer (engineering S. and T.); and **Mr. R. C. Rattray,** executive officer (engineering development) to the Railway Executive.

Mr. R. E. Ellis has been appointed general manager of British Polar Engines Ltd., 139 Helen Street, Glasgow, S.W.1.

Mr. G. A. St. C. Emery has been appointed chief engineer of Philidas Ltd.

Mr. J. H. B. Forster has been appointed Chairman and **Mr. B. Chetwynd Talbot** deputy chairman and managing director of South Durham Steel and Iron Co., Ltd., and Cargo Fleet Iron Co., Ltd.

Sir John B. Greaves has been appointed a director of Ruston & Hornsby Ltd.

Mr. C. K. F. Hague, M.I.Mech.E., A.M.I.E.E., managing director of Babcock & Wilcox Ltd., has been elected President of the British Engineers' Association.

Mr. Roland Hudson has been appointed London manager of John Holroyd & Co., Ltd.

Col. D. R. Johnson, M.S.M.A., M.I. Ex., M.E.B.R.A., has resigned his appointment as Sales and Advertisement Manager with the Pitter Gauge and Precision Tool Co. Ltd.

Mr. S. A. Lane has been appointed a director and general manager of the Brush Electrical Engineering Co., Ltd.

Mr. Charles Lawrie has been appointed an additional general sales manager of the Vacuum Oil Co., Ltd.

Mr. I. A. Marriott has been appointed managing director of W. G. Bagnall, Ltd.

Mr. H. P. R. Scott has been appointed joint managing director of Thomas Summerson & Sons, Ltd., of Darlington, and of Summerson's Foundries Ltd.

Rear-Admiral M. S. Slattery has joined the Board of Short Brothers and Harland, Ltd.

Dr. R. L. Smith-Rose, M.I.E.E., has been promoted to the new post of Director of Radio Research in the Department of Scientific and Industrial Research.

Mr. J. M. Surrall has been appointed a director of Simplex Electric Co., Ltd., a subsidiary company of Tube Investments Ltd.

Lieut.-Commander G. W. Wells has been appointed joint general manager; **Mr. A. Robinson,** technical consultant and director; **Engineer Rear-Admiral C. W. Lambert,** general works manager; **Mr. A. Jackson, Mr. G. D. Elliott** and **Mr. W. Geary,** works managers; **Mr. K. Paterson,** chief mechanical engineer; and **Mr. J. L. Gaskell,** chief electrical engineer, of the Appleby-Frodingham branch of The United Steel Companies, Ltd.

Mr. G. F. Wix has been appointed a director of the Phosphor Bronze Co., Ltd., Witton-Birmingham.

BUSINESS NOTES.

Adcock and Shipley Ltd., Leicester. A. C. Wickman Ltd. announce their acceptance of the sole agency for the products of Messrs. Adcock and Shipley Ltd., in their Glasgow and Newcastle sales areas.

These areas cover the whole of Scotland and the counties of Northumberland, Cumberland and Durham, as well as the northern parts of Westmorland and the North Riding of Yorkshire.

Adcock and Shipley build a well-established line of milling machines and are also specialists in the manufacture of special-purpose machinery built to order from customers' own design.

The British Electricity Authority announces that its headquarters are as from February 1st, at British Electricity House, Great Portland Street, London, W.1. Telephone: Museum 6844.

World's Most Elaborate Radio-telephone System for New Oil-Pipe Line. Recognition of Britain's top-ranking position in radio engineering is underlined by the award of a £300,000 contract for one of the most elaborate telephone and radio communication systems yet devised. Radio beams will span the whole length of a new oil-pipe line which is to be laid between the Persian Gulf and a Mediterranean port 800 miles distant.

The order has been given to **The General Electric Co., Ltd.,** of England, whose scheme was chosen from among several British and American tenders. When finished it will embody the combined skill of the expert in every known branch of telephone and radio engineering.

The equipment will include the main V.H.F. beamed transmitting and receiving stations; high, medium and low power H.F. transmitting and receiving stations; unattended automatic radio repeater stations; telephone exchanges; mobile H.F. transmitting and receiving stations and "walkie-talkies." Transmitting stations will use both amplitude and the new frequency systems of modulation.

The complete system will be the most advanced of its type in the world and will provide an excellent example of the application to peaceful purposes of Very-High-Frequency communication technique developed during wartime by the G.E.C. Research Laboratories at Wembley, London.

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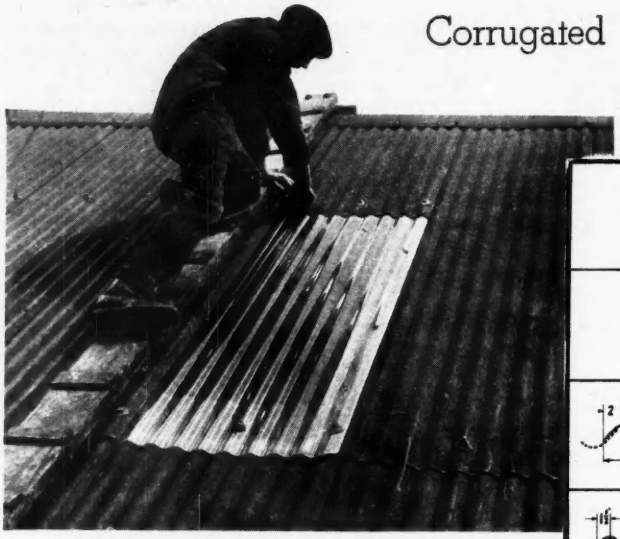

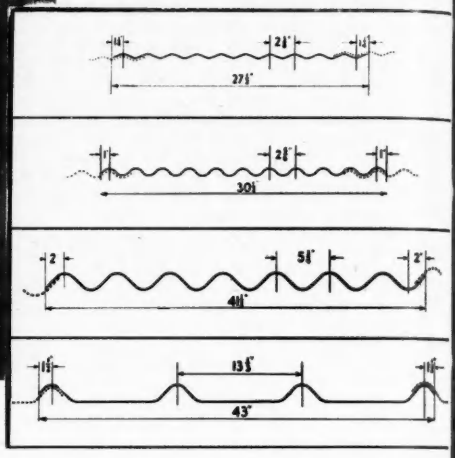

Corrugated "Perspex" acrylic sheet can be fixed as easily as normal corrugated sheet, using ordinary hook bolts and washers. Made in nine different profiles, and available in lengths from 3 ft. to 7 ft. in intervals of 6 ins. For full particulars write to :—

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MANCHESTER. *Phone: BLA. 9181.*

The Record Electrical Co., Ltd., of Broadheath, Altrincham, Cheshire, announce the removal of their Glasgow office to new and more convenient premises at 162 Buchanan Street, Glasgow, C.1. Telephone: Douglas 4205. Mr. R. G. Dean is in charge of the office.

The Chloride Electrical Storage Co., Ltd., have received an order from the Anglo-Iranian Oil Co., Ltd., for a number of standard Exide 6-volt batteries for use in conjuction with seismic recorders on refraction survey work.

It appears that the underground rock structure in any particular area, previously selected by geologists as likely to disclose an oil pool, can be investigated before drilling is started. This is done by creating an artificial earthquake by setting off a buried explosive charge.

Industrial Correspondents Group Visit A. C. Wickmans' Coventry Factories. On Friday, 23rd January, some thirty members of the Industrial Correspondents Group, which included representatives of the National Press and B.B.C. visited the Coventry factories of A. C. Wickman Ltd. A quick tour of each of the three factories showed the visitors production methods adopted for the manufacture of the firm's well-known line of high production machine tools, and they witnessed demonstration of these machines producing components on machines ready tooled for despatch to customers' works.

Visiting the Works of Powderloys Ltd., they saw the production processes required in the manufacture of Wimet Powders and also the great care exercised in quality control. Following this tour, a visit was made to the Wimet Tool Works where the visitors were able to see the comparative performance achieved with high speed steel and Wimet Tungsten Carbide Tools on the same job.

A problem which the Americans had been unable to solve themselves was the manufacture of dies for the production of Gillette Razor Blades. As a result of the use of Wimet as a tool material for the production of these blades, 3,000,000 blades could now be produced with the one die before resharpening became necessary. Previously, only 200,000 had been produced between regrinds.

Perhaps the high spot of the tour was the demonstration of the Cornelius Electronic Comparator when various applications of this unique instrument were described and demonstrated.

ADDRESS BY CHAIRMAN OF THE AUTOMOBILE DIVISION.

At the General Meeting of the Automobile Division of The Institution of Mechanical Engineers, held on January 6th at Storey's Gate, St. James's Park, London, S.W.1, the recently-elected Chairman of the Division, Capt. G. T. Smith-Clarke, M.I.Mech.E. (Technical Director of Alvis Ltd.) delivered his Address. This was largely biographical in character and was of especial interest to automobile engineers as it described the author's improvisatory difficulties in the early days of the Industry when he was serving his apprenticeship in the Road Motor Dept. of the Great Western Railway. Capt. Smith-Clarke also gave a graphic description of his development work on aero engines with Alvis Ltd. during the two world wars, in addition to design progress on the Alvis car.

DORMER INFORMATION BOOKLET

This booklet produced by **The Sheffield Twist Drill & Steel Co., Ltd.,** is primarily to provide engineers, draughtsmen, designers, estimators, machine tool operatives, tool room workers, buyers, etc., not only with a list of stocked sizes of Twist Drills manufactured, but with technical information of the utmost value, the whole being presented in a clear concise manner. The booklet will be supplied free of charge.

BOOKS RECEIVED.

Garcke's Manual of Electrical Undertakings and Directory of Officials. Edited by Frederick C. Garrett. Publishers: Electrical Press Ltd., 23 Great Queen Street, London, W.C.2, 1196 pp. Price: £3 3s. The latest edition, Volume 45, of Garcke's Manual for 1947-48 has just been published. It follows the line of previous editions and contains the financial, engineering and supply statistics of all electricity undertakings in Great Britain and Ireland. This year's edition also contains the latest particulars available at the time of going to press of the British Electricity Authority and the Area Boards, which will provide the new organization for the supply of electricity under national ownership. Section II. gives details of Manufacturing and Allied Companies. An up-to-date Directory of Directors and Officials connected with the Electrical Industry is also included.

Proceedings of the Society for Experimental Stress Analysis, Vol. V. No. 1. Edited by C. Lipson and W. M. Murray. Publishers: Addison-Wesley Press, Inc., Kendall Square, Cambridge 42, Mass., U.S.A., 136 pp. Price: $6.00. The latest volume of this valuable collection of articles contains the following important papers:—Explosive Impact Tests by Paul R. Shepler; Criterion of Static and Fatigue Failures by Albert Sniderman; The Experimental Determination of the Dynamic Structural Response of an Airplane to Impact Loadings by W. M. Claflin; A Proposed New Shock Measuring Instrument by W. P. Welch; Radio Pulse Telemetering by Conrad H. Hoepner; Method of Obtaining the Stress at the Mid-Thickness by Measurements from Only One Surface of a Plate by A. Boodberg and E. D. Howe; Bonded Wire Strain Gage Techniques for Polymethyl Methacrylate Plastics by A. G. H. Dietz and W. H. Campbell; Effect of Temperature Gradients upon the Introduction of Residual Stresses in Weldments or other Structures by J. K. Stitt; Recent Contributions to the X-Ray Method in the Field of Stress Analysis by J. T. Norton and D. Rosenthal; Lateral Vibration and Stress in a Beam under Shock Machine Loading by E. G. Fisher; High-Temperature Strain Gages and their Application to Measurement of Vibratory Stresses in Turbosupercharger Buckets by S. S. Manson, R. H. Kemp and W. C. Morgan; Some Characteristics of Navy "High Impact" Type Shock Machines by Irwin Vigness; Electronic Commutation of Strain Gages for Telemetering by Lawrence Lee Rauch; and Determination of the Effect of Ground Impact Forces in the Airplane Drop-Test by A. H. Pedersen and J. G. MacCarthy.

The 1947 edition of the "S.I.M.A. Handbook" (Scientific Instrument Manufacturers' Association of Great Britain Ltd.) has just been published by the Scientific Instrument Manufacturers' Association of Great Britain. The main feature of this 220-page book is an alphabetical index (containing about 2,000 entries) of the products made by the members of the Association, supplemented by a list of these firms with their products. By using these lists, it is easy to discover in a few moments the names and addresses of the makers of almost any type of scientific instrument, as well as the products of any particular firm. Copies may be obtained from the Association at 26 Russell Square, London, W.C.1, at 10s. 6d., post free.

NEW EQUIPMENT.

Plessey Centrifugal Switches. A range of centrifugal switches has been developed by the Plessey Company Ltd., of Ilford, suitable for 2-4 or 6 pole 50 cycle motors up to 1½ h.p.

The switches are available in two sizes, large and small, the small one being suitable for motors up to ¾ h.p., ⅝ in. dia. shaft and the large for motors up to 1½ h.p., and $1^{1}/_{16}$ in. dia. shaft.

Each switch consists of two parts, one being mounted on the rotor and the other on the stator end shield. When the motor attains some 72-82 per cent of synchronous speed the rotor portion expands radially and, by engaging with the outer tongue of the stator portion, opens the switch and disconnects the motor starting winding. When the motor is switched off and slows down, the contraction of the rotor re-sets the switch for the next start.

These switches are now in production and are described in an illustrated brochure just published.

The Hymatic Engineering Co., Ltd., Redditch, Worcestershire, has developed a new Air Hammer which can be operated from any shop air line or compressor. While weighing no more than 20 oz., it has an air consumption of 5 c.f.m. at 80 p.s.i. delivery. Its simple trigger action gives one-handed, finger-tip control for either a featherweight touch or a full strength blow (air pressure being 50/150 p.s.i.). The butt is shaped for a comfortable grip and an adjustable cover diverts the cold exhaust air from the operator's hand. Also a quick acting chuck with octagonal socket prevents the tool from falling out or rotating unintentionally.

Three different kits of tools can be supplied—"Building," "Metalworking" and "Woodworking"—each one comprising the essential tools for the respective jobs.

Classified Advertisements.

The Rate for all classified advertisements is 6d. per word; in bold print 9d. per word; minimum order 6s. Box number advertisements 1s. extra. Instructions together with remittance must be received not later than the 3rd of each month for advertisements to appear in the same month's issue.

AGENCIES

H. FRANK WHITE, 20 years in engineering sales and purchasing, useful connections London and overseas trade, invites correspondence from medium size manufacturer, view to agency or development work. Personal attention, convenient office in Bank district. 2 Laurence Pountney Hill, London, E.C.4.

MACHINERY, ETC., FOR SALE

ALUMINIUM AND ALUMINIUM ALLOY SHEETS, 10-30g., also bars, tubes, and sections; brass strip and rods; copper sheet and strip. Immediate delivery from stock.—**Almex Ltd.,** Imperial Works, Watery Lane, B'ham, 9.

AIR COMPRESSORS up to 30 c.f.m., electric motor driven and spray painting equipment for immediate delivery from manufacturers.—**Alfred Bullows & Sons, Ltd.** (Dept. 1), Long Street, Walsall, Staffs., England.

NISSEN TYPE HUTS, Ex-Government Stock, 36 ft. by 16 ft., £70 and £80; 24 ft. by 16 ft., £56 and £45. Other Buildings also available. Write for details.—Box, E.D., **Universal Supplies,** 45 Pickford Lane, Bexleyheath, Kent. Telephone: Bexleyheath 4227. Yard and Depot: Crabtree Manorway, Belvedere, Kent.

To the Advertisement Manager, "THE ENGINEERS' DIGEST," 120, WIGMORE STREET, LONDON, W.1.

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